DESIGN, DEVELOPMENT, AND EVALUATION OF A HEAT EXCHANGER FOR A LINEAR FIXED MIRROR SOLAR CONCENTRATOR

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DESIGN, DEVELOPMENT AND EVALUATION OF A HEAT EXCHANGER FOR A LINEAR, FIXED MIRROR SOLAR CONCENTRATOR

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NOMENCLATURE

h_f	convective front heat transfer coefficient
V	wind velocity
h_{fr}	radiant heat transfer coefficient from front side of heat exchanger
q_{fr}	radiative net heat loss from the front side of the heat exchanger
A_c	area of the front of the heat exchanger
T_c	temperature of front cover
T_s	temperature of ambient air
ε_c	emissivity of front cover
σ	Boltzmann constant
h_b	forced convection back side heat transfer coefficient
D_o	outer cover diameter
k_a	thermal conductivity of fluid
ν_a	viscosity of fluid
C	constant defined in equation (5)
n	constant defined in equation (5)
Re_D	Reynolds number
h_c	free convection back side heat transfer coefficient
Gr_D	Grashof number
Pr	Prandtl number
h_{br}	radiant heat transfer coefficient from back side of heat exchanger
ε_b	emissivity of back side
T_b	temperature of back side
q_ℓ	total heat loss from the heat exchanger

NOMENCLATURE (Concluded)

A_b	area of back side
q_u	useful heat input into pipe air flow
q_k	total heat to reach the heat exchanger
η	collection efficiency
T_p	temperature of inside pipe

SUMMARY

The purpose of this project was to design, develop and evaluate a heat exchanger for a linear, fixed mirror solar concentrator. The design utilized economical, readily available materials and required no special assembly techniques. The final design of an eight-foot section consisted of two steel pipes, two stainless steel reflecting strips, insulation, and a durable transparent cover.

Taking into consideration all important heat transfer processes the heat exchanger efficiency was assessed.

Testing was performed over an existing 560 square foot linear solar concentrator on the SSTC #1 Building at Georgia Tech. Data were collected under steady state conditions. The air and collector surface temperatures were measured at several locations along the heat exchanger for various flow rates. The results of the experimental performance were then compared with those theoretically determined.

CHAPTER I

INTRODUCTION

Background and History

More than twelve centuries have passed since the Greek historian Galen (130-220 AD) recorded an account of the siege of Syracuse by the Romans in 212 B.C. During this siege, the Greek Scientist, Archimedes (287-212 B.C.) succeeded in setting fire to Roman warships by means of a "burning mirror" [1]. Such an account seemingly gives the first record of the practical application of focusing solar collectors.

All energy producing techniques face periods of trial and error before a proper formula for their utilization is found. Now is such a period for the development of solar energy, but since the days of Archimedes many applications for the sun's heat energy have been found. The accomplishments in solar research range from simple solar heaters and ovens to solar homes, heat pumps and large solar furnaces.

The best type of device used to collect solar energy depends primarily on the application. Flat plate thermal energy collectors are used for heating water and buildings but can provide temperatures of only about 150 °F above ambient. If higher temperatures are desired the sunlight must be concentrated onto the collecting surface [2].

For any solar collecting unit with the exception of furnaces the heat exchanger plays a necessary role. A solar heat exchanger, as opposed to heat exchangers which transfer heat from fluid to fluid, transfers the thermal energy of the sun's rays into some heat exchange fluid.

This fluid may be water or air or any of a wide variety of other heat transfer fluids. The choice depends upon the application. Solar heat exchangers may be used with or without concentrators. Flat-plate collectors, which are themselves solar heat exchangers according to the previous definition, utilize both direct and indirect solar radiation. The area of the solar energy absorber is the same as the area which intercepts the solar radiation, i.e. the aperture area. Heat exchangers used with solar concentrators have an absorber area smaller than the concentrator area. Focusing collectors, or concentrators, increase the energy flux on the solar energy absorber; this arrangement is most practical for processes where high temperatures are required.

For both types of heat exchangers the primary components are the "black" absorbing surface, the cover plate over the surface which is transparent to solar radiation, and the insulation on the back side.

Definition and Purpose of Research

This project involved the design, development, and evaluation of a heat exchanger for a linear, fixed mirror solar concentrator. The purpose of the research was to compare the experimental performance of the heat exchanger with what was theoretically expected. Comparisons were made over a range of flow rates.

It was decided that the heat exchanger be a round duct with air serving as the heat exchange media. The duct was designed and built to near optimum conditions (smooth inner surface, geometrically symmetrical, etc.) so that in performing similar experiments correlations with these experimental results might be obtained. The model was built with the additional fact in mind that similar models might be commercially available which would stress the importance of ease of fabrication. No attempt was made to design and construct a heat exchanger which would exhibit the maximum performance characteristics possible without regard to cost.

The model was tested with a linear solar concentrator of the fixed mirror type requiring that the heat exchanger be moved in order to remain in focus throughout the day. Data were collected under steady state conditions.

Method of Attack

The project was divided into three areas:

(1) design and construction of the heat exchanger;

(2) mounting of the equipment and experimental testing of the exchanger; and

(3) theoretical analysis of the exchanger.

The first step took into consideration the principles of modern solar air heaters. Since the heat exchanger was to be cost effective, the design involved low cost readily available materials.

The second step included mounting of the finished model over an existing linear solar concentrator on the campus of the Georgia Institute of Technology with subsequent experimental evaluation. Data were collected under steady state conditions and temperatures were recorded at several locations in and on the heat exchanger for various flow rates.

Finally, taking into consideration all important heat transfer processes, an analysis was performed to predict the performance of the heat exchanger. The results of the analysis were compared with experimental data.

CHAPTER II

THE CONCEPTS OF HEAT EXCHANGER DESIGN

General Considerations

For solar air heat exchangers, the variables of consideration are much the same as those of a conventional solar air heater. It was mentioned in the previous section that important aspects of the heat exchanger design are the "black" absorbing surface, the covers which are transparent to solar radiation, and the back side insulation.

The purpose of "blacking" the absorber plate is to increase the absorptivity of the plate, so that the plate will absorb more thermal energy than it would have if left "unblackened". Often used to "blacken" absorber plates are selective coatings [3] which can significantly improve the performance of solar heaters by increasing the collector efficiency. However, due to the difficulty of producing selective surfaces flat-black painted absorbers are still predominatly used.

Insulation around the sides of the absorbing unit which are not exposed to the useful solar radiation is necessary to keep unwanted heat losses by conduction to a minimum.

As for the cover material, the type and how many layers are considered while the spectral transmittance properties are examined. The cover material should have a high transmissivity for radiation transmission and low reflectivity to maintain a minimal amount of reflected radiation. Since a good "greenhouse" effect is desired, the material should be relatively opaque to infrared radiation from the absorbing plate. In addition the composition of the chosen material should provide durability if the cover is to be subjected to increased solar intensity or many cycles of the same.

In general, the higher the temperature required, the more covers used; the principle underlying the use of multicovers being each air layer between two successive covers provides a transparent insulation against heat losses from the collecting plate to the atmosphere. The temperature of the outermost cover panes becomes progressively lower with increases in the number of cover panes, and hence the heat losses from the outermost pane to the atmosphere are reduced. However, with a layer number of cover panes, the reflective losses increase (in addition to the cost) so that unless very high temperatures are desired, it is not the practice to employ more than two cover panes [4].

A design factor that is concomitant with the number of cover panes is the stagnant air gap thickness. A stagnant air gap interposes a high impedance to convective heat flow between the absorber plate and the ambient air. The gap limits the mechanisms of heat transfer to free convection and radiation. The losses, both of radiation and convection can be reduced to low values by the use of multiple covers (mentioned above) but the consequent reduction in transmission of solar radiation makes more than one air gap of doubtful value.

Design of the Heat Exchanger Model

With a more detailed description of the solar concentrator to follow, it will suffice here to mention that the concentrator was composed of ten 8-foot sections. The optical design was in accordance with the geometry described in Reference [5] (see Figure 1). Each section consisted of 28 mirror slats each 2.9 inches wide which were positioned symmetrically with 14 on each side of the array centerline.

In effect the model will be serving as an air heater as well as an air duct. Cobble [6] analyzed and performed heat balances on two collectors for the parabolic cylinder type of concentrator, one being a flat plate heat exchanger, the other a cylindrical heat exchanger. He reported that the flat plate heat exchanger was the optimum choice for this type of concentrator while the cylindrical heat exchanger was more a choice of convenience.

The convenience factor played the major role in the decision that the duct be cylindrical. Round ducts are moderately inexpensive and are readily available if elaborate materials are not required.

To conform with the recommendations mentioned previously, the round pipe was blackened to increase the absorptivity of the receiving surface. Another note, worthy of mention, is that the receiving surface of the pipe would be only that area exposed to the concentrator. This area, roughly one-half of the total area, is all that is required to be "blackened". The back side of the pipe could be left in the same condition as purchased.

Because of the angular spread of 32.2 minutes of a beam of reflected sunlight (due to the finite diameter of the sun) the focused image on the focal plane will not have the same width as the mirror slats but rather, the 2.9 inch width slats will reflect the sun's beam to a width of 3.65 inches at a distance of 100 inches from the slats. The 100-inch distance is the diameter of the concentrator's reference circle

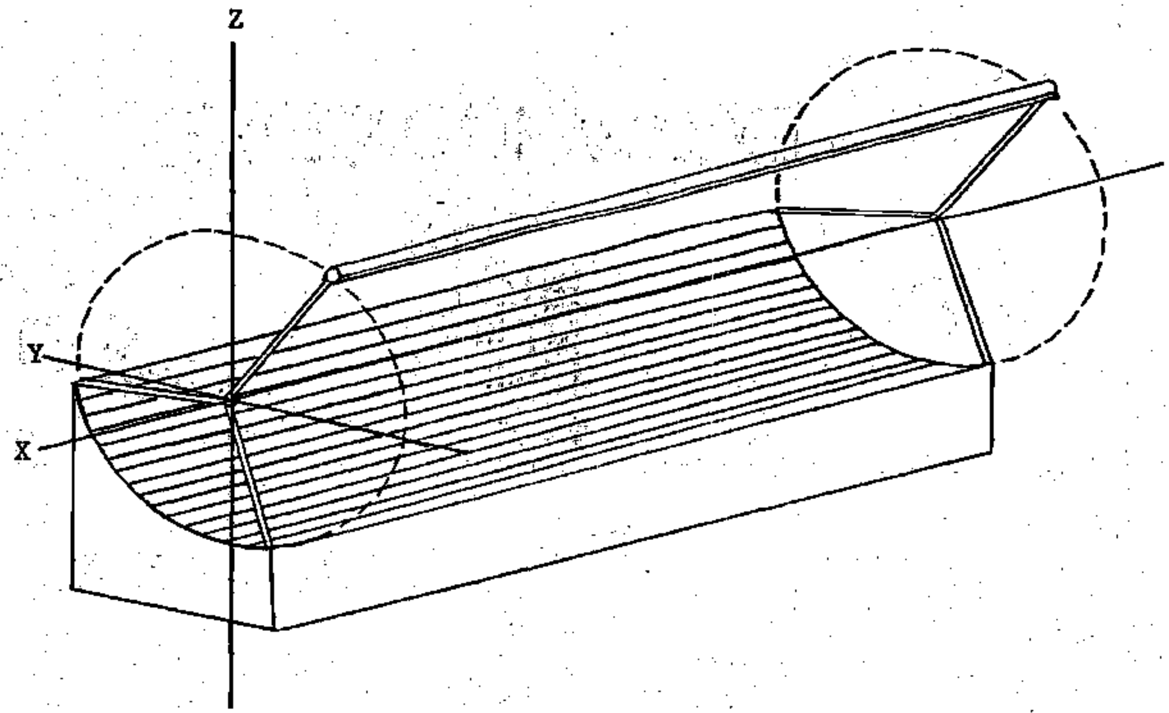

Figure 1. General View of Concentrator with Heat Exchanger and Supports

and is the location of the line at which the mirrored slats have been brought into focus. Note that the 3.65 inch width of the focal line is really the upper bound calculation. The mirror slat would have to be perfectly perpendicular to the sun's rays to expose the entire 2.9 inch width to the sun, and only at certain times of the day will certain mirror slats be truly perpendicular to the sun's rays. Since the heat exchanger would be directly in line with the sun and the certain perpendicular mirrors, a shadow would be cast on those mirrors making the shadowed portion virtually useless. Thus, the 3.65 inch width of the focal line is truly an upper bound.

Rather than employ a 3.65 inch diameter pipe to serve as the heat receiving tube, it may be better to use a smaller diameter tube and employ the use of a terminal concentrator on the sides of the tube. The terminal concentrator is a flat reflecting surface attached to each side of the receiving pipe (see Figure 2). The frontal opening of the terminal concentrator is at least 3.65 inches. The purpose of this concentrator is to reflect the bordering edge of the focal line onto the pipe, yielding extra concentration onto the receiving surface, in turn leading to higher temperatures of the surface. The advantage of having a smaller pipe is two-fold. For good heat transfer in any air heater the ratio of the duct diameter to the length of the duct should be as small as possible while still allowing satisfaction of air flow requirements. The size of the pipe is also significant when considering the size of the shadow that the heat exchanger will cast over the concentrator at certain times of the day. Since a concentrator of the type to be used utilizes only direct radiation, any shadowed area of the concentrator will be use-

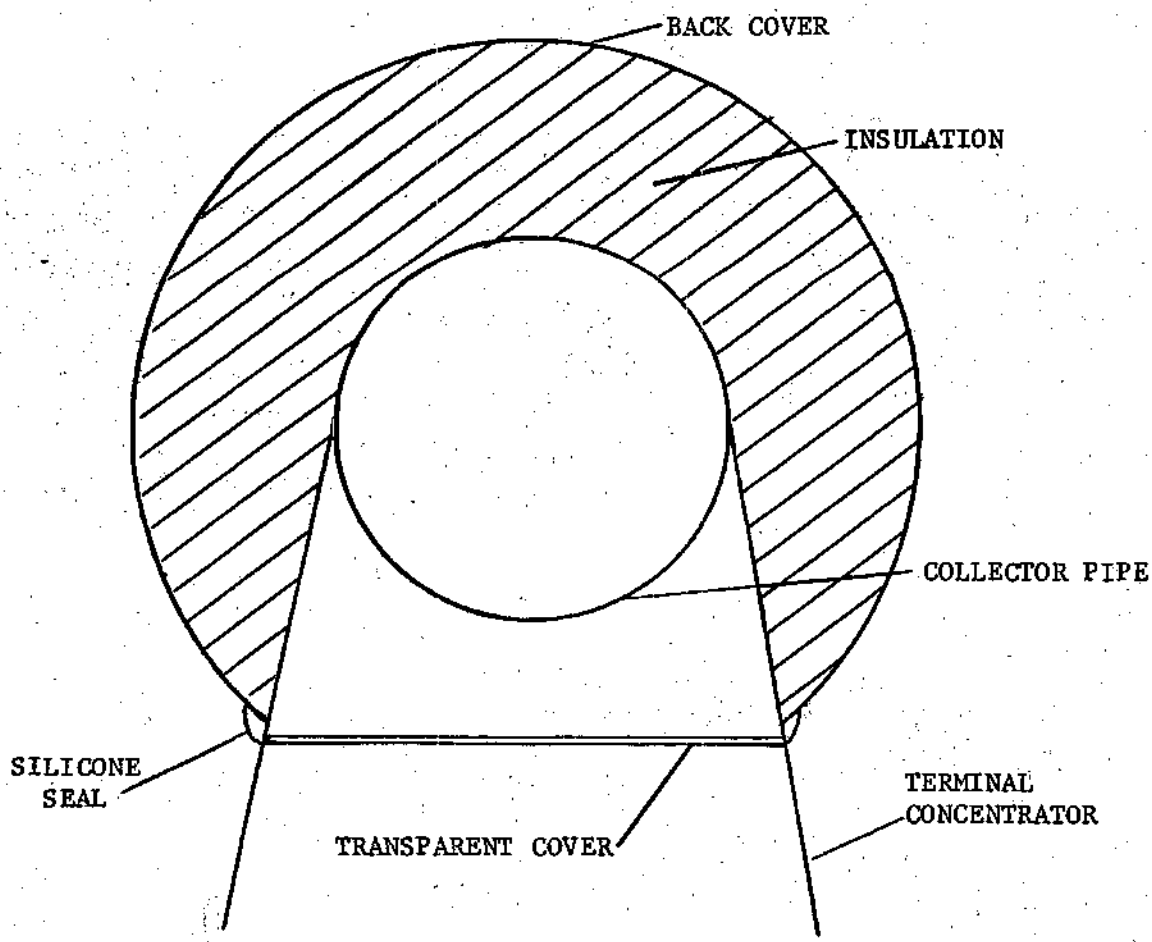

Figure 2. Cross Sectional View of Heat Exchanger

less. Thus, the width of the receiving pipe is significant and the smaller it is the smaller the shadow.

With the terminal concnetrator to be placed on the side of the pipe the attachment of a cover pane is simple. While the choice for cover panes would be made from transparent panes they need to be self-supporting and flat (example: window pane glass). If the panes are self-supporting, they would be stiff enough to add structural support to the terminal concentrator to prevent any random expansion or contraction of the frontal opening of the concentrator. The flat covers give the best aforementioned static support and are more readily available than covers of any other geometry.

The gap between the receiving surface of the pipe and the cover surface serves as the stagnant air gap which, as mentioned earlier, minimizes the conduction and convection heat losses from the front of the collecting surface.

Insulation is placed over the portion of the receiving pipe which is not receiving the radiation from the concentrator. The insulation (not too thick as to lose the shadowing advantage obtained by choosing the smaller pipe) is effective enough to keep conduction heat losses through the back of the pipe to a minimum. Since insulation chosen is not waterproof, a thin cover is placed over the insulation to make it, as well as the entire heat exchanger, weatherproof.

As described in the previous section, the receiver duct is a circular pipe as a matter of convenience and availability. A primary consideration of pipe material is its conductivity. Intuitively, it would be expected the better pipe would be the one that would most evenly distribute the heat collected around the circumference of the pipe. It would not be reasonable for the air heated from the receiving portion of the tube to heat the back portion of the pipe. From practical considerations the choice of pipe material was narrowed down to aluminum and steel. Copper was eliminated due to its higher cost. However, since the surface temperature of the pipe could reach about 1000 °F, the decision was made against the use of aluminum since the melting point of aluminum is not much higher. Steel allows a larger safety margin. The thermal conductivity of steel, though only 20% that of aluminum is high enough to be considered for this application.

Keeping in mind that a terminal concentrator is employed, the decision was made to use a three-inch diameter pipe. Steel pipes of three-inch diameter and .0156 inch thickness are a standard size of stove pipe which are readily available and inexpensive. For even distribution of the heat about the tube circumference a thicker walled pipe would be more suitable than this choice of pipe. However, up to now, no mention was made of the weight limitations of the heat exchanger. The design of the concentrator called for heat exchanger supports to be located every eight feet along the length of the concentrator but the aluminum framework was not designed to support a heat exchanger in excess of 30 pounds per eight-foot length. The chosen tube proved not only to be cost-effective but lightweight enough to satisfy the heat exchanger weight limitation.

Upon examination of many brands of commercial black paints, stove-black with its high temperature characteristics was chosen to coat the receiving portion of the pipe. The paint was applied to only one-half of the pipe which is roughly the area of the pipe exposed to the rays from the concentrator.

For the choice of reflecting surfaces for the terminal concentrator, the decision was made against using mirrors due to their fragility and the forseeable trouble which would have been encountered trying to construct the model. It was decided to use thin strips of highly polished stainless steel. The polished surface produces a fairly high reflectivity and the use of the metal lends to the ease of fabrication of the heat exchanger model. The strips can be easily fastened to the sides of the receiving pipe with short metal screws. The opening of the terminal concentrator on the receiver needs to be at least 3.65 inches. The decision to make the frontal width four inches allows for slight misalignment of any of the mirror slats, and also for increased beam spreading at large sun angles.

Common window glass was considered as the economic choice of transparent cover, but the composition of the glass provides a thermal expansion which does not allow it to withstand repeated thermal stresses which result from being in the focal line of the concentrator. Pyrex was chosen instead because of its high temperature stability and low thermal expansion which minimizes thermal stresses. Pyrex is the trademark for a brand of glassware manufactured by Corning Glass Works, Inc.

The insulation used around the back of the pipe was chosen to be Fiberfrax. Fiberfrax is the trademark for a type of insulation manufactured by the Carborundum Company. Fiberfrax was obtained in blanket form which was ideal for applying and shaping to the contour of the back of the pipe. Common fiberglass insulation would have met this same criterion but the thermal conductivity is roughly twice as high as Fiberfrax and it is much less refractory.

Since the Fiberfrax insulation is not weatherproof, another steel pipe of sufficient diameter was cut lengthways along the seam and spread to fit snugly over the insulation. The result would then be an outer cover of cylindrical shape (see Figure 2) thus providing cheap weather-proofing of the insulation and adding to the heat exchanger's outward attractiveness.

Since supports for the heat exchanger were located every eight feet, it was convenient to construct the heat exchanger in eight-foot lengths. The steel tubes were available in standard lengths of ten feet but were cut to eight feet. The interlocking end of the inner pipe was left intact for section joining.

The dimensions of the stainless steel for the terminal concentrator were eight feet by four inches. Each strip was 25 mils thick.

The brand of stove-black paint was manufactured by Triple-X Chemical Laboratories, Inc.

The Pyrex pane dimensions were 47.5 inches by 4 inches. It required two of these panes to cover one heat exchanger section. The one-inch gap was required for the heat exchanger support to mount onto the inner tube surface.

The Fiberfrax was purchased in blanket rolls of 15-inch width and .75 inches thick.

The outer pipe to be split along the seam to overlay the pipe was chosen to have a five-inch diameter. Each edge of the split seam met the outside of the terminal concentrator along the length of the heat exchanger (see Figure 2). A seam of silicone rubber sealant was applied to each side of the terminal concentrator where the outer pipe came into contact in order to weatherproof the heat exchanger.

The Pyrex panes were held into place by small metal screws placed at specific locations through the terminal concentrator sheets. The Pyrex rested on these screws while the pressure encountered as the outer cover presses against the sides of the terminal concentrator prevented any movement of the panes.

CHAPTER III

THEORETICAL ANALYSIS

General Considerations

The purpose of the analysis here is to predict the collection efficiency of a heat exchanger of the aforementioned type by observing the heat losses. Assumptions are made in the analysis which simulate the model in operation. The experimental data are consulted for accurate temperature estimation when the need for such estimation arises.

It was implied previously that the heat exchanger is basically a solar air heater. Duffie [7] outlines several simplifying assumptions for solar air heater heat loss calculations. Important assumptions are:

(1) performance is steady-state;

(2) there is one dimensional heat flow through the tube walls, insulation, and Pyrex cover;

(3) there is a negligible temperature drop through the tube walls and Pyrex cover;

(4) properties are independent of temperature;

(5) heat loss through front and back are to the same ambient temperature; and

(6) heat loss from the terminal concentrator will be neglected.

Heat loss from the heat exchanger can be divided into two catagories: heat loss from the portion of the model facing the concentrator and heat loss from the portion of the model unexposed to the concentrator. The exposed portion includes the area subtended by the Pyrex cover and will be referred to as the front of the heat exchanger. The unexposed portion is the entire area covered by the outer pipe and will be referred to as the back of the heat exchanger.

For calculation of the heat loss from the front of the exchanger due to convection effects McAdams [8] suggests

$$h_f = 1.00 + .669V \tag{1}$$

where h_f is the convective wind heat transfer coefficient and is in Btu/hr sq. ft. F. The wind velocity, V, is in feet per second. The empirical equation is restricted to wind flow over flat plates. His formulation is such that a convective coefficient will exist if the wind velocity is zero, thus allowance is made for free convective effects.

To account for the radiative heat losses from the front of the heat exchanger to the surroundings a radiant heat transfer coefficient will be employed. This radiant coefficient, h_{fr}, while similar to the convective heat transfer coefficient will cause the rate of heat flow to become linearly dependent on the temperature difference and can be incorporated directly into a thermal network. Define the radiant heat transfer coefficient for the front cover, h_{fr}, as

$$h_{fr} = \frac{q_{fr}}{A_c\ (T_c - T_s)} \tag{2}$$

where q_{fr} is the radiative net heat loss from the Pyrex cover, A_c is the frontal area of the Pyrex and $(T_c - T_s)$ is the temperature difference between the cover and the surroundings.

Since the radiative net heat loss, q_{fr}, from the cover is

$$q_{fr} = \varepsilon_c A_c \sigma \; (T_c{}^4 - T_s{}^4) \tag{3}$$

where ε_c is the emissivity of the cover and σ is the Boltzmann constant the radiant coefficient becomes

$$h_{fr} = \varepsilon_c \; \sigma \; (T_c{}^2 + T_s{}^2) \; (T_c + T_s) \tag{4}$$

Radiative heat loss from the blackened pipe surface which is transmitted through the Pyrex cover will be neglected along with any conduction heat losses from the cover.

For forced convection losses from the outside back of the heat exchanger, Hilpert's [9] results will be used. He accurately measured the conductances for air flowing over cylinders of diameters as large as six inches. His empirical equation,

$$\frac{h_b \; D_o}{k_a} = C \left(\frac{V \; D_o}{\nu_a} \right)^n \tag{5}$$

is for calculation of the average heat transfer coefficient, h_b, of a circular cylinder in a fluid flowing normal to its axis. D_o is the cylinder diameter, V is the wind velocity, and k_a and ν_a are the conductivities and viscosities of the fluid respectively. The constants, C and n, are dependent upon the Reynolds number, Re_D, where

$$Re_D = \frac{V\ D_o}{\nu_a} \tag{6}$$

Table 1 gives the coefficients for Hilpert's equation within various ranges of Reynolds numbers.

Table 1. Coefficients for Hilpert's Equation

Re_D	C	n
0.4 - 4	0.891	0.330
4 - 40	0.821	0.385
40 - 4,000	0.615	0.466
4,000 - 40,000	0.174	0.618
40,000 - 400,000	0.0239	0.805

An equation for the average heat transfer coefficient from a horizontal pipe in free convection, recommended by McAdams [8] on the basis of experimental data is

$$\frac{h_c\ D_o}{k_a} = 0.53\ (Gr_D\ Pr)^{.25} \tag{7}$$

where Gr_D and Pr are the Grashof and Prandtl numbers of the fluid, respectively.

The radiant heat transfer coefficient for the back of the heat exchanger is given by h_{br}, where

$$h_{br} = \varepsilon_b \, \sigma \, (T_b^{\,2} + T_s^{\,2}) \; (T_b + T_s) \qquad (8)$$

where ε_b is the emissivity of the back surface and T_b is the temperature of the back surface. This radiant coefficient is formulated in the manner used to obtain the radiant coefficient for the front of the exchanger.

Now all the coefficients in the previous heat loss equations can be incorporated to find the total heat loss, q_ℓ, from the heat exchanger where

$$q_\ell = A_c(h_f + h_{fr}) \; (T_c - T_s) + A_b(h_b + h_c + h_{br}) \; (T_b - T_s) \qquad (9)$$

A_c and A_b represent the area of the outside areas of the Pyrex and back side covers, respectively.

The useful heat q_u is given by the heat balance equation for the heat exchanger where

$$q_u = q_k - q_\ell \qquad (10)$$

and q_k represents the total heat to reach the heat exchanger.

The collection efficiency, η, is defined as

$$\eta = \frac{q_u}{q_k} \tag{11}$$

Now it can be seen that the primary performance measurement can be calculated by being able to accurately predict the amount of heat loss from the heat exchanger.

Analysis

In the following analysis the heat loss equations from the previous section are employed to calculate the collection efficiency of the heat exchanger. In the calculations necessary to obtain such heat loss data, the experimental data were consulted to accurately predict the outside surface temperatures of the heat exchanger as a function of pipe temperature. Figures 3 and 4 show the best line obtained through the experimental data measurements of the pipe temperature, T_p, versus the Pyrex cover temperature, T_c, and the temperature of the back side of cover, T_b. The data points shown were collected at an air flow rate of four cubic feet per minute. One of the reasons for undertaking the heat loss analysis as a function of pipe temperature is because the mass flow rate of air through the pipe never enters into the analysis until the final efficiency calculations.

On the days that the experimental measurements were taken, the wind was usually gusting between three and five miles per hour. For correlation of theoretical to experimental analysis, a wind speed of four miles per hour was assumed the wind speed in the theoretical calculations. In addition, four miles per hour would not be an unrealistic estimation

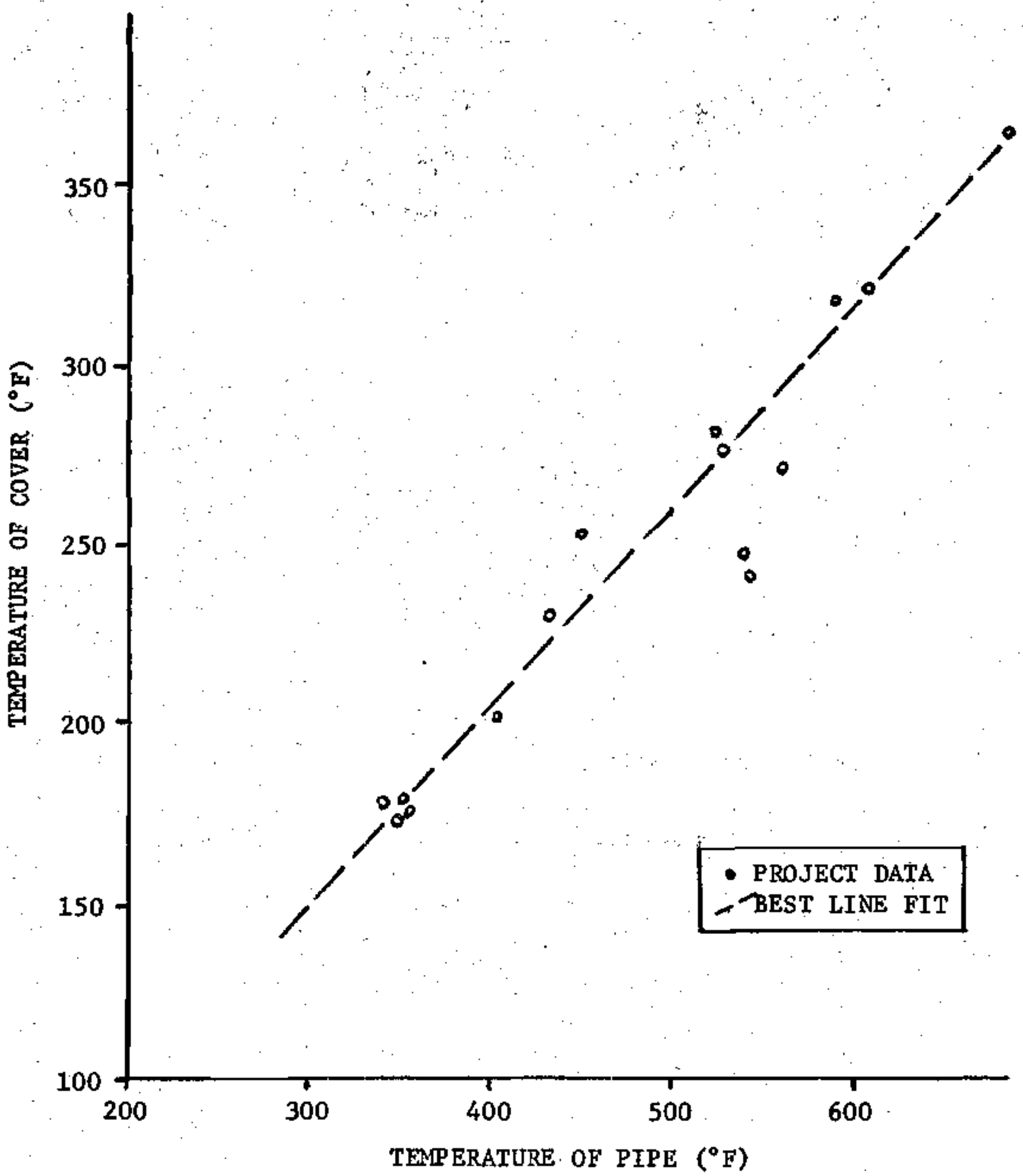

Figure 3. Temperature of Pipe Versus Temperature of Pyrex Cover at Flow Rate of 4cfm Through the Pipe

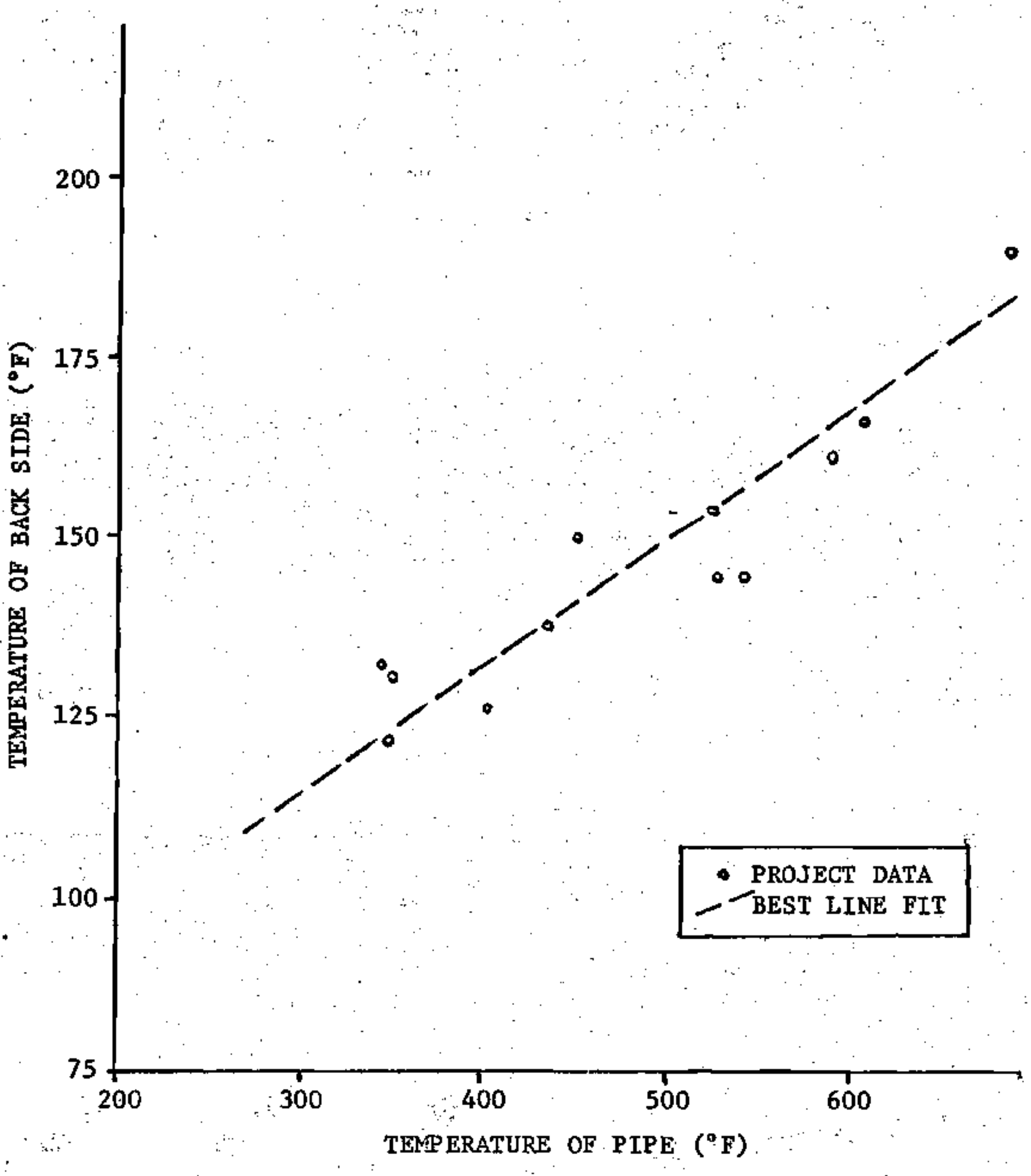

Figure 4. Temperature of Pipe Versus Temperature of Back Side of Heat Exchanger at Flow Rate of 4 cfm Through the Pipe

if practical conditions are considered. At this wind speed, however, free convection effects on the back of the heat exchanger could be neglected. A preliminary comparison showed that at most the free convection coefficient was one-fifth the amount of the forced convection coefficient. However, the free convection term is temperature-difference dependent and the calculation was carried out at a temperature difference too extreme for normal operating conditions. Under normal operating conditions the free convection term would be less than one-fifth with wind velocity remaining at four miles per hour.

To complete the theoretical heat loss calculations, the following values were assumed:

(1) T_s is 80 °F

(2) ε_c is .88

(3) ε_b is .23

Using the above assumptions along with the corresponding surface temperature estimations, Figure 5 is obtained which shows the heat losses from the model as a function of pipe temperature. It may suffice to point out again that the wind velocity was assumed to be four miles per hour.

With total heat loss per linear foot on hand, the experimental data are again used for the amount of incident direct radiation which falls on the concentrator. This data was gathered with a pyrheliometer. The best lines falling through these data points are shown in Figures 6-9 for flow rates of 4, 8, 12, and 16 cubic feet per minute. Again pipe temperature is plotted along the abscissa.

With the assumption of 15 percent edge losses on the concentrator and 15 percent reflection losses assumed and accounted for, Figure 10 shows the theoretical efficiencies at the aforementioned flow rates as a function of pipe temperature.

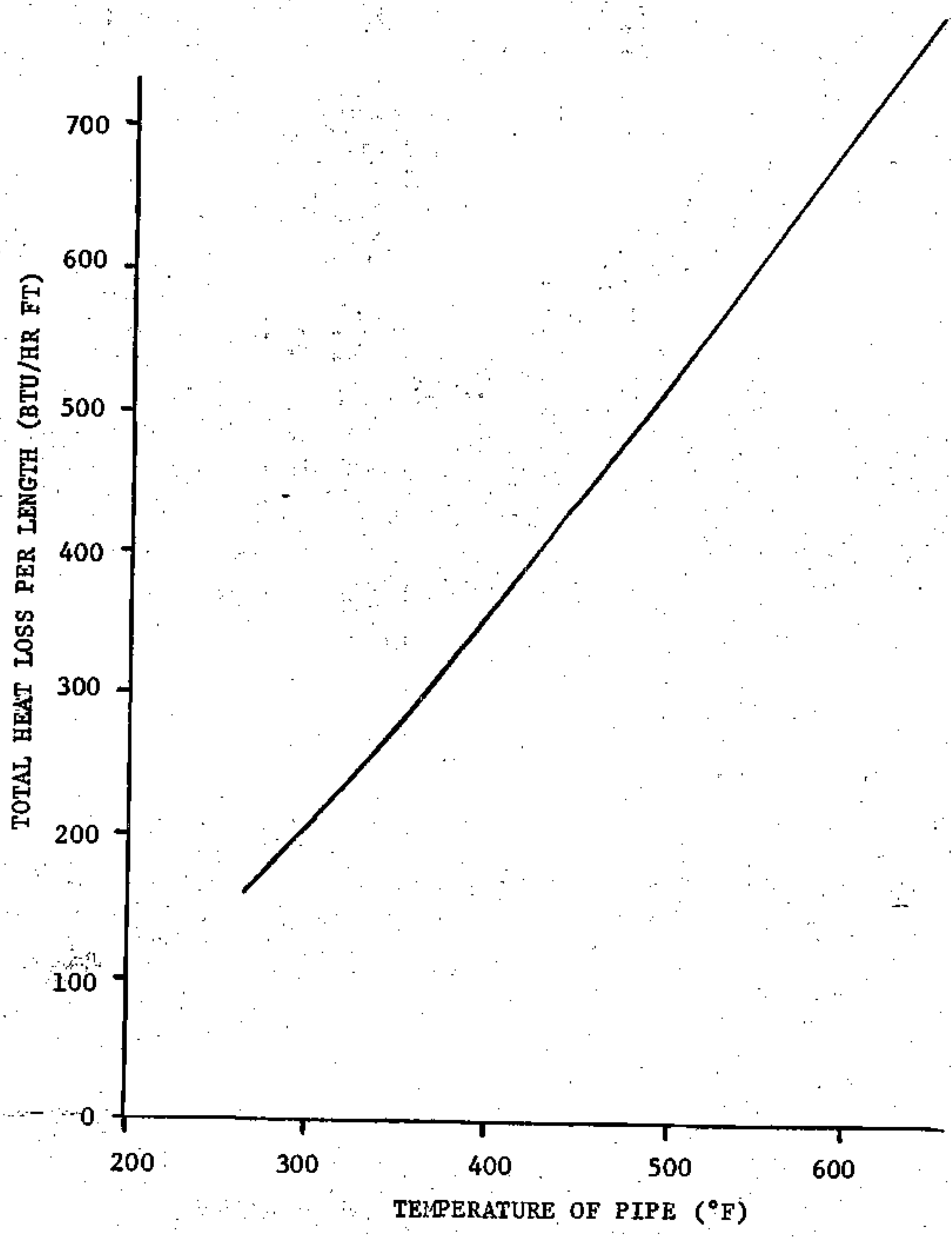

Figure 5. Temperature of Pipe Versus Total Heat Loss per Linear Foot of Heat Exchanger at Wind Velocity of 4 mph

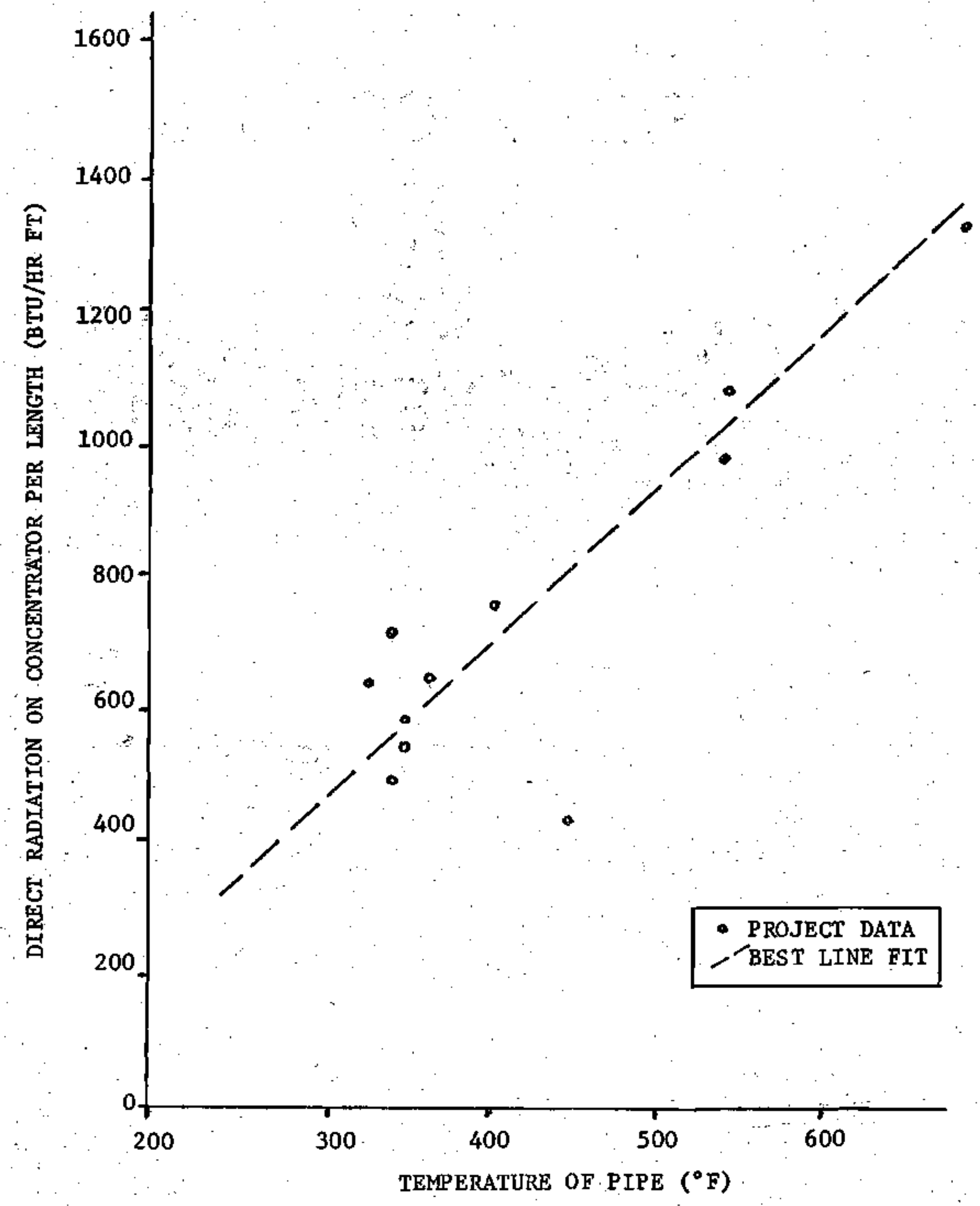

Figure 6. Temperature of Pipe Versus Direct Radiation on Concentrator at Flow Rate of 4 cfm Through the Pipe

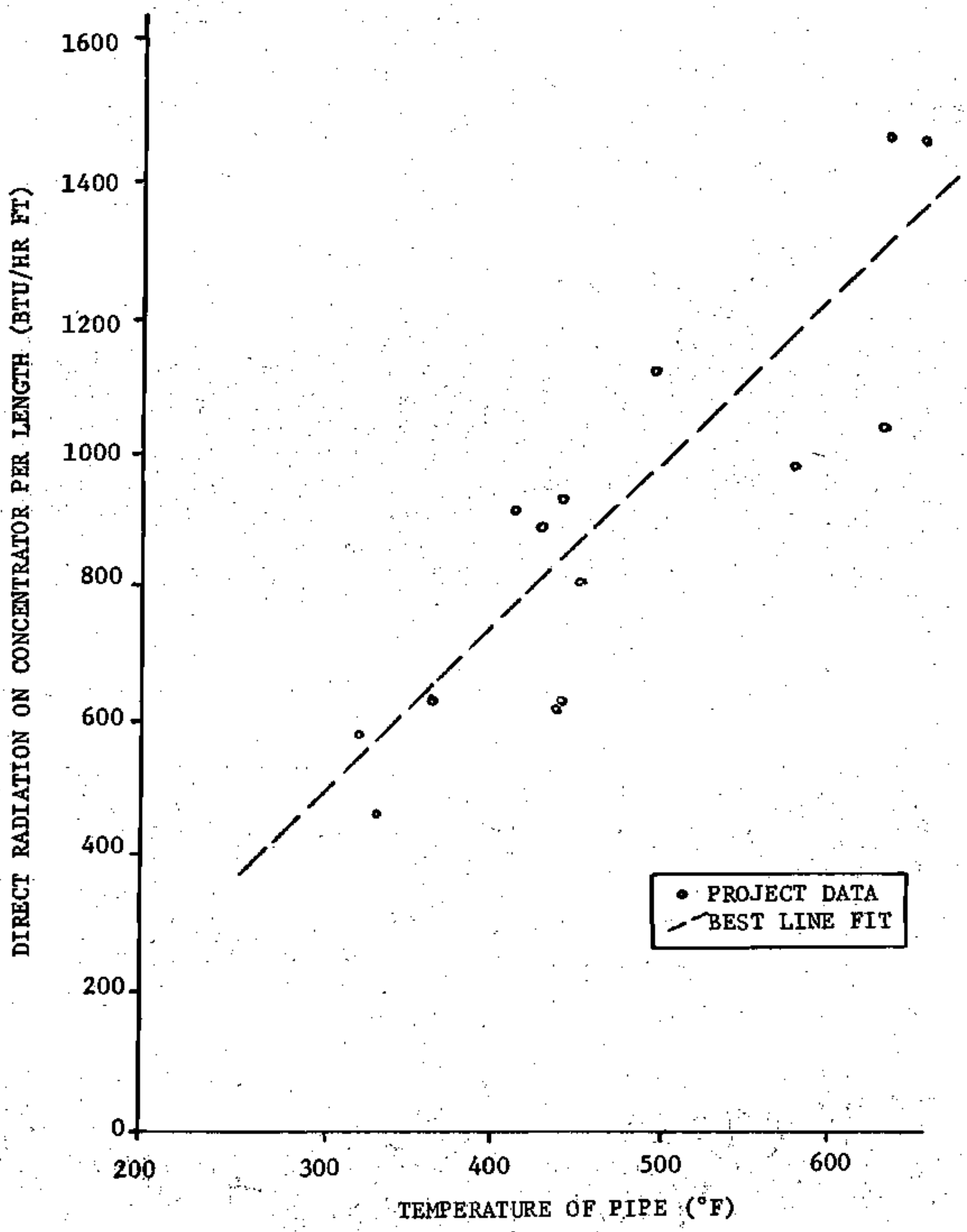

Figure 7. Temperature of Pipe Versus Direct Radiation on Concentrator at Flow Rate of 8 cfm Through the Pipe

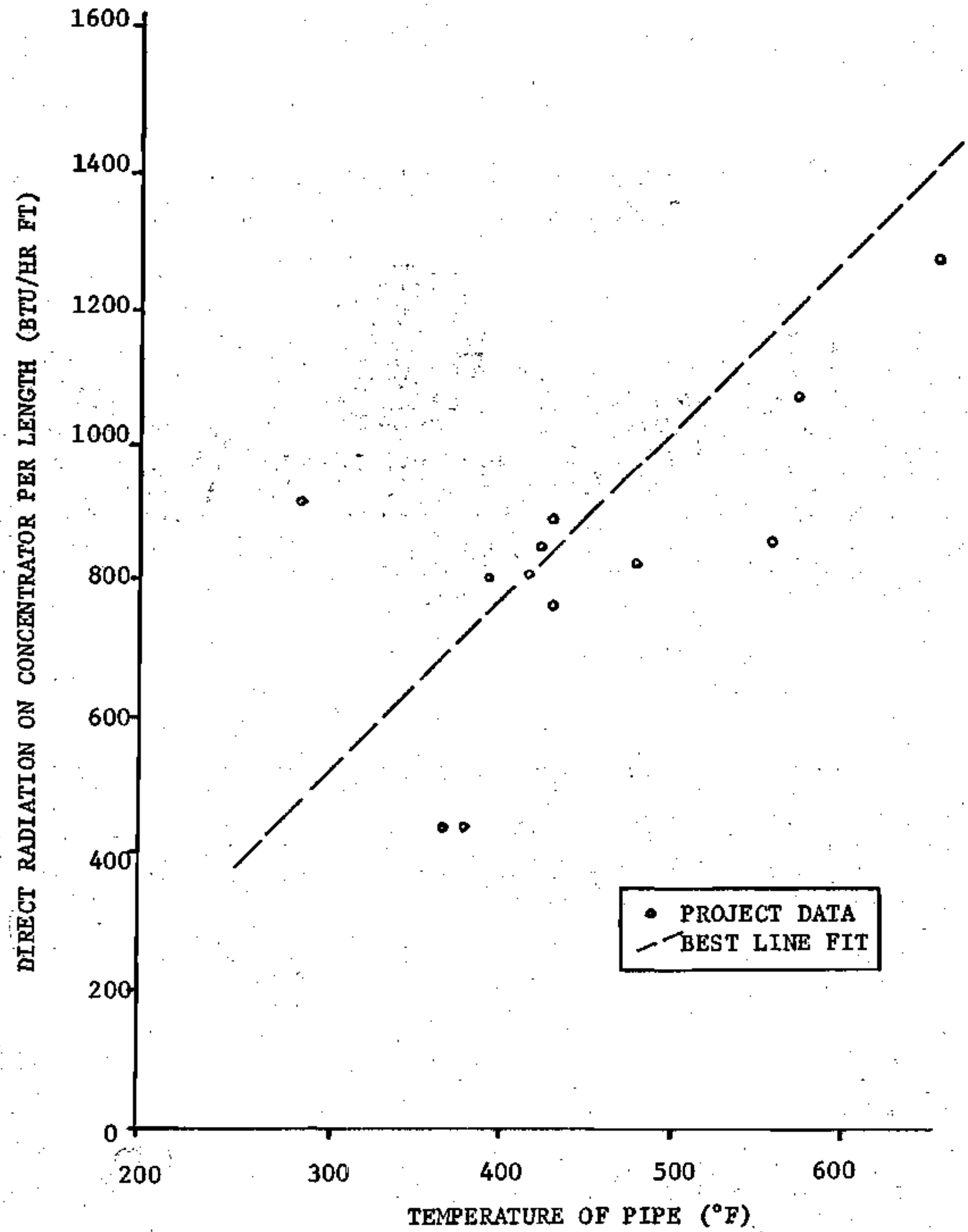

Figure 8. Temperature of Pipe Versus Direct Radiation on Concentrator at Flow Rate of 12 cfm Through the Pipe

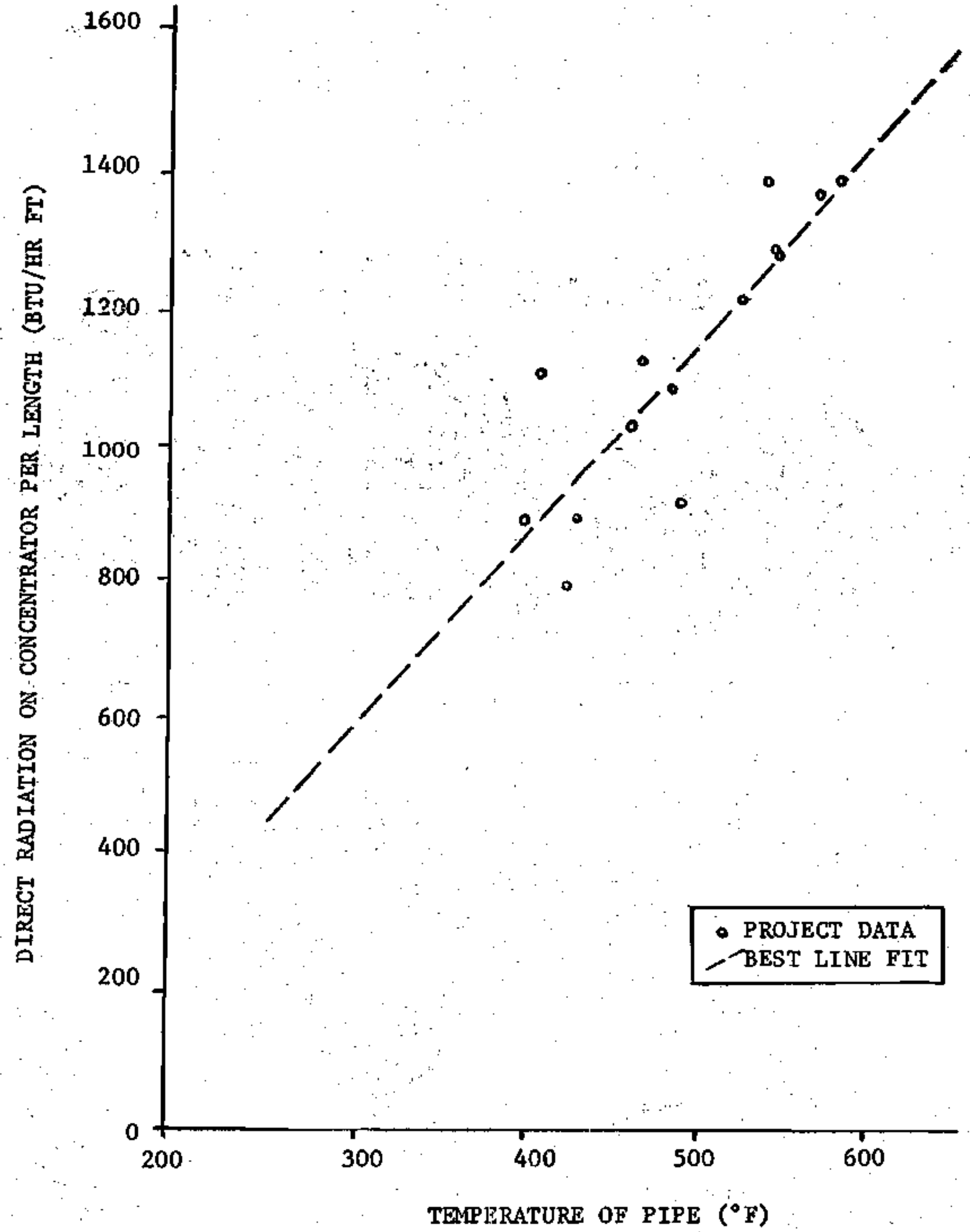

Figure 9. Temperature of Pipe Versus Direct Radiation on Concentrator at Flow Rate of 16 cfm Through the Pipe

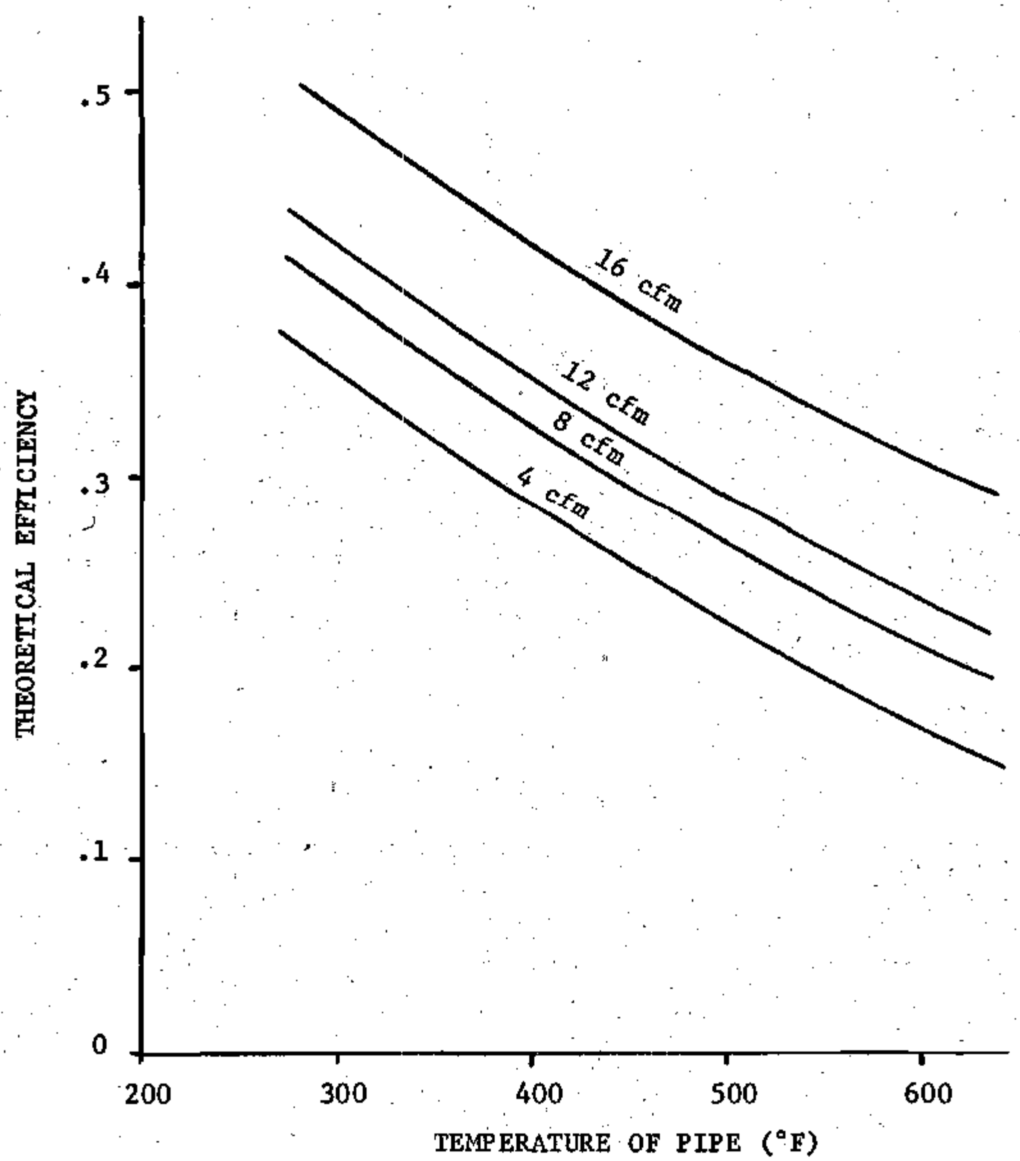

Figure 10. Temperature of Pipe Versus Theoretical Efficiency at Various Flow Rates and Wind Velocity at 4 mph

CHAPTER IV

EXPERIMENTAL PROCEDURE

Experimental Apparatus

The heat exchanger model was tested in the focal line of a FFMC solar concentrator. The optical design was in accordance with the geometry described in Reference [5]. The reflecting array was composed of 28 mirror slats each having a width of 2.9 inches and which were positioned symmetrically with 14 on each side of the array centerline. The included angle of the reflecting system was 105° (see Figure 11). One-hundred inches was the diameter of the reference cylinder.

The concentrator was designed for east-west orientation. The array was tilted toward the southern horizon so the tilt angle (see Figure 11) was equal to the latitude of the installation site. For Atlanta, Georgia, the tilt angle is 33.7 degrees.

The solar concentrator components were fabricated by Scientific Atlanta, Inc. of Atlanta, Georgia, and assembled at Georgia Tech. It was about 7 feet wide and 80 feet long, composed of ten 8-foot sections. Much care was taken in the design of the mirror supports to eliminate the need for mechanical adjusting of each individual slat. The latter was accomplished by designing an aluminum rib for each end of the 8-foot extrusions which was cut to hold all of the 28 extrusions in a specified position. Positioning of the extrusions was accomplished with sufficient accuracy to insure that not more than ten percent of the reflected light misses the theoretical illuminated area.

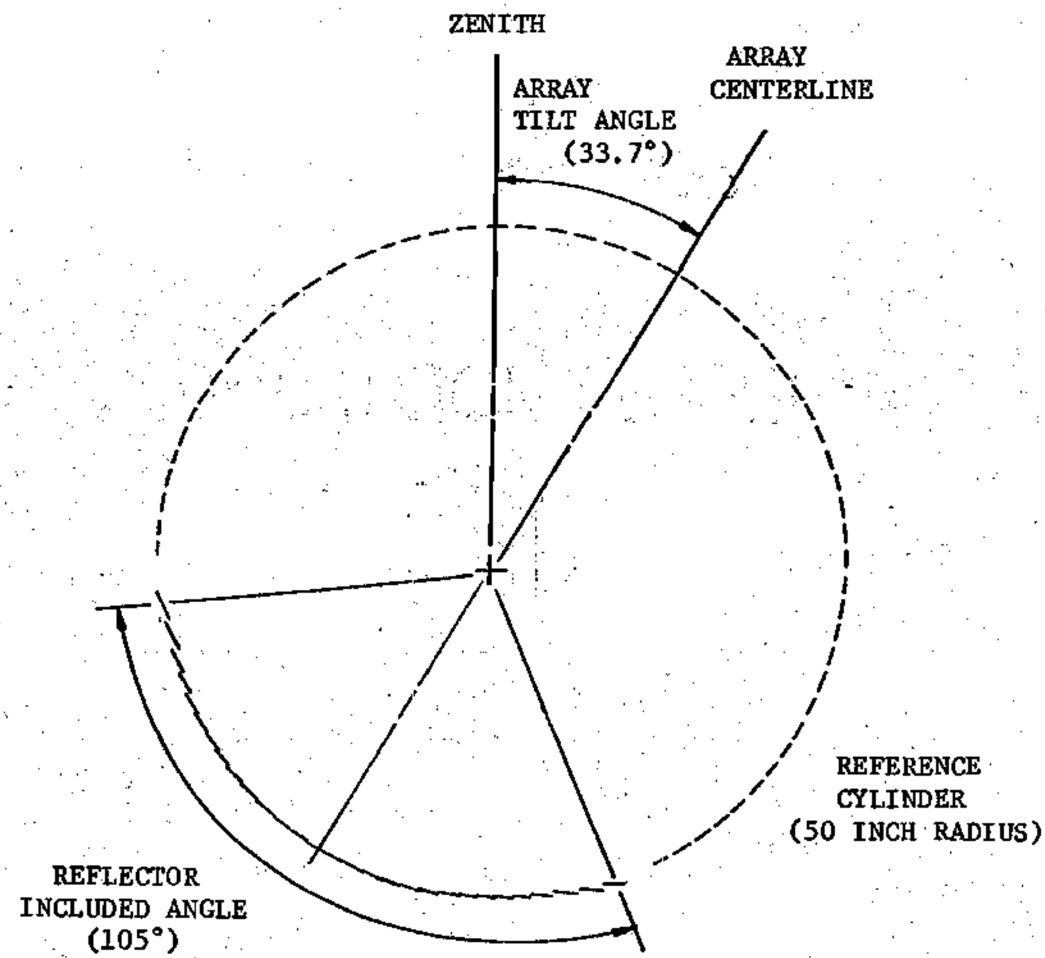

Figure 11. Concentrator Geometry with View Along X-Axis

The supports for the heat exchanger were spaced at eight-foot intervals. The support system was designed that the front face of the heat exchanger would always be facing the center of the reflecting array.

Since the concentrator is of the fixed mirror type, the heat exchanger was designed to move and stay in the focus of the concentrator as the sun moved across the sky. An automatic sun tracking unit was employed to keep the exchanger in the focal line. The heat exchanger was moved into focus by an electric motor drive system. The tracking unit was complete with an automatic timing clock for turning the system on and off but manual overrides were provided for all operations.

Since the concentrator was in eight-foot sections and the supports for the heat exchanger were located at eight-foot intervals it was convenient to analyze the heat exchanger in eight-foot sections. The findings would be valid for any of the eight-foot sections.

Also included in the list of experimental apparatus is the blower used to force air through the heat exchanger. The blower was of the centrifugal type and was rated at one-half horsepowerand 27 volts dc.

A flowmeter was attached to the exit of the blower and could accurately measure the flow rate of air up to 18 cfm. It was calibrated at an air temperature of 70 °F and 14.7 psia.

The many temperature measurements were taken by employing iron-constantan thermocouples and the leads were connected to a temperature potentiometer.

Wind measurements were made on an instrument which gave instantaneous wind speeds and direction. Direct insolation measurements were taken on an Eppley pyrheliometer and instantaneous values were recorded at two-minute intervals.

Data Collection

For evaluation of the air temperature rise through the pipe a total of five thermocouples were used. Five small holes were drilled in the top of the pipe to allow for the insertion of the wires which were lowered to the center of the pipe. The thermocouple wire was of number 24 guage which would resist significant movement even in high air flow rates. The thermocouples were spaced at two-foot intervals.

To observe the temperature distribution around the inside pipe, five thermocouples were spaced and welded along the circumference at 45° intervals around one side of the pipe. The first was located at the center of the blackened surface and the last was at the center of the pipe back side. Each of these thermocouples were along the circumference of the pipe at the center.

To observe the difference in pipe surface temperature along its linear length, two more thermocouples were welded to the center of the blackened surface. Each of these were located three feet from the center of the pipe. Two thermocouples were placed in the stagnant air layer between the transparent cover and the collecting surface. Each of these were located about 2.5 feet from the center of the model.

Two thermocouples were glued to the outside of the transparent cover with a high-temperature silicone rubber sealant. Each of these were located about two feet from the center of the pipe along the centerline of the cover. Two more thermocouples were welded to the outside cover along the center of the back side, each located about 2.5 feet from the center of the pipe. An additional thermocouple was welded to the terminal concentrator near the center of the pipe in the stagnant air layer. The last thermocouple of the total of 20 employed was left unattached from the potentiometer in order to measure the ambient air temperature.

For recording purposes the thermocouples were coded with a number. Table 2 lists the number of the thermocouple with its specific location along the heat exchanger. Figure 13 schematically shows the locations.

Table 2. Location of Thermocouples

Number	Location
1	center of air flow, at exit end of pipe
2	center of air flow, 2 feet from exit
3	center of air flow, center of pipe
4	center of air flow, 2 feet from entrance
5	center of air flow, at entrance end of pipe
6	center of blackened surface, 1 foot from entrance
7	center of blackened surface, center of pipe
8	45° around pipe from no. 7
9	90° around pipe from no. 7
10	135° around pipe from no. 7
11	180° around pipe from no. 7
12	center of blackened surface, 1 foot from exit
13	on terminal concentrator at center of pipe in stagnant air gap
14	center of stagnant air gap, 1.5 feet from exit
15	center of stagnant air gap, 1.5 feet from entrance
16	center of transparent cover, 2 feet from exit
17	center of transparent cover, 2 feet from entrance
18	center of cover back side, 1.5 feet from exit
19	center of cover back side, 1.5 feet from entrance
20	in ambient air

Data collection was made on four separate days. The days were

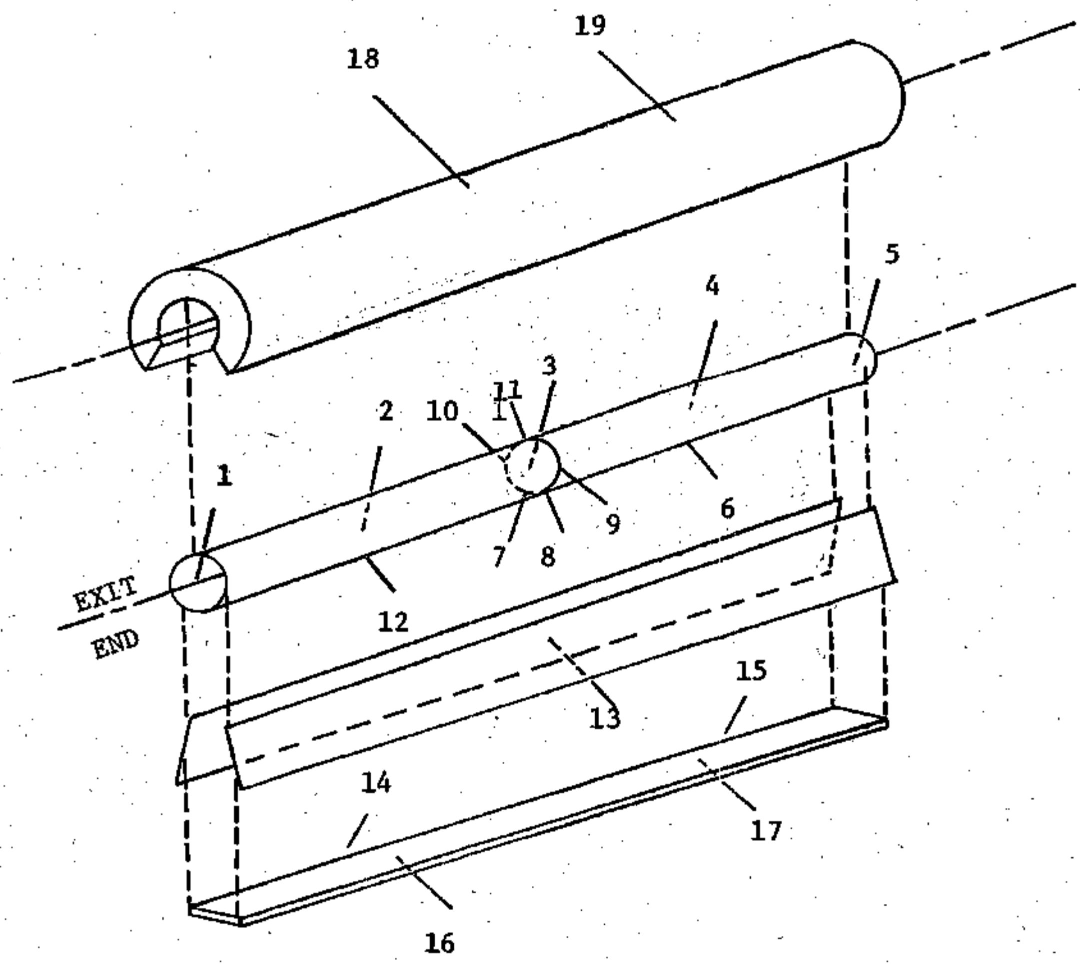

Figure 12. Location of Thermocouples

May 30, June 3, June 4, and June 5, 1975. The days were clear with scattered cumulus clouds. Winds were gusting up to five miles per hour.

Figure 3 shows the experimental measurements of the average pipe temperature versus the average temperature of the Pyrex cover for the four days at a flow rate of 4 cubic feet per minute through the pipe. The dotted line is the best fit line (obtained by the method of least squares) through the points. Measurements taken at the flow rates of 8, 12, and 16 cfm could also have been plotted on the same figure but were omitted to avoid clutter. Such plots would have showed the near identical trend of measurements at different flow rates. Figure 4 shows the experimental measurements of the average pipe temperature versus the average temperature of the back side of the heat exchanger. Again, only the points collected at a flow rate of 4 cubic feet per minute are shown.

The average temperature of the pipe was obtained by averaging temperature measurements taken at station numbers 7, 8, 9, 10, and 11. This was justifiable since the temperature measurements taken at stations 6, 7, and 12 (the front side of the pipe) were consistently linear with respect to each other and the stations 7 through 11 were situated about the circumference of the center. The Pyrex cover average temperature was obtained by averaging measurements at stations 16 and 17. The average temperature of the back of the heat exchanger was obtained by averaging station measurements 18 and 19.

Figures 13-16 show the measured temperature difference between stations 2 and 5. Station 1 data was discarded when it was discovered upon disassembly of the model that the No. 1 thermocouple wires had been in contact near the upper wall of the pipe. Such contact led to erroneous

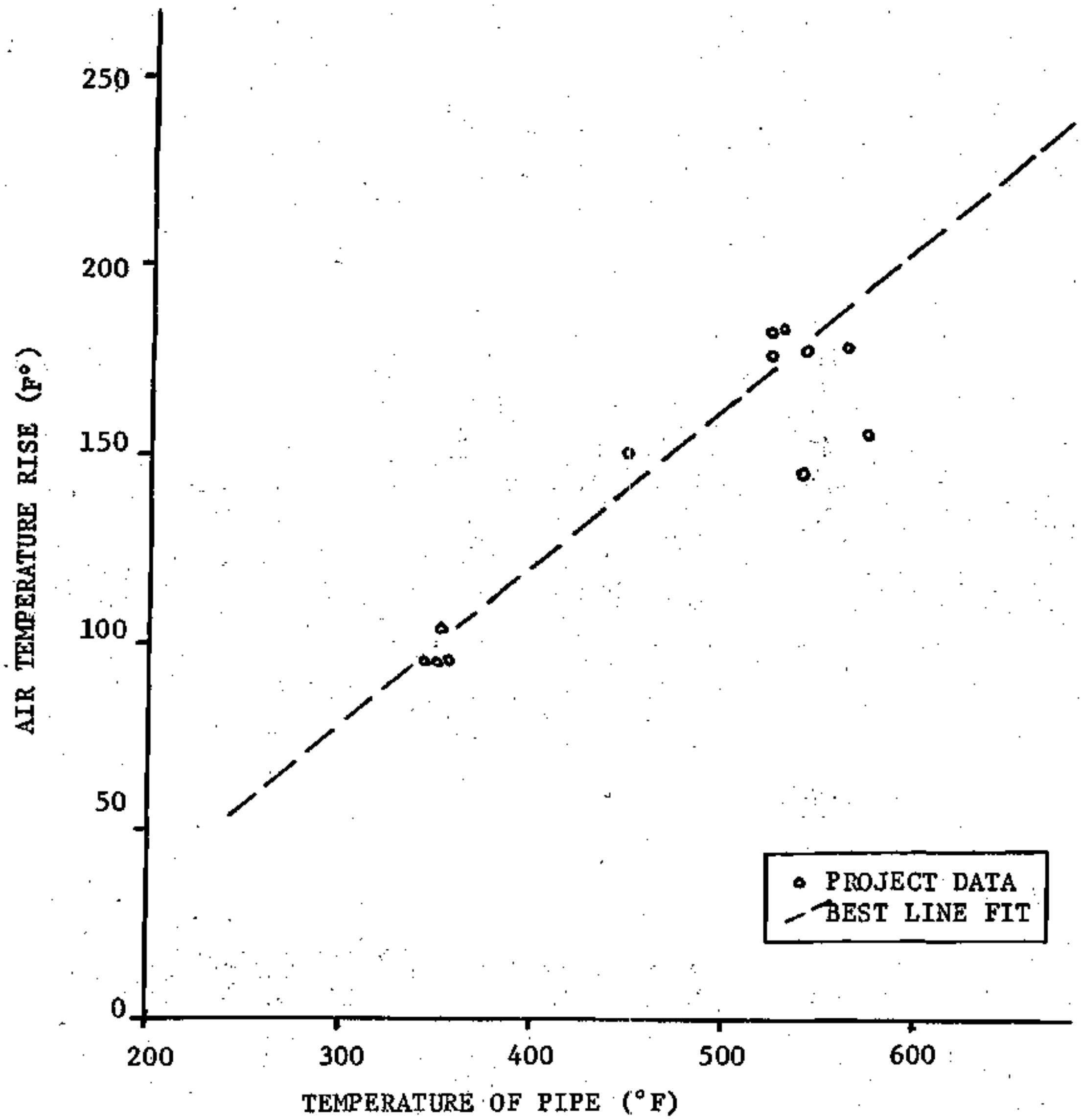

Figure 13. Temperature of Pipe Versus Temperature Rise of Air Through 6 Linear Feet of Heat Exchanger at Flow Rate of 4 cfm (Average Temperature of Inlet Air was 340 °F)

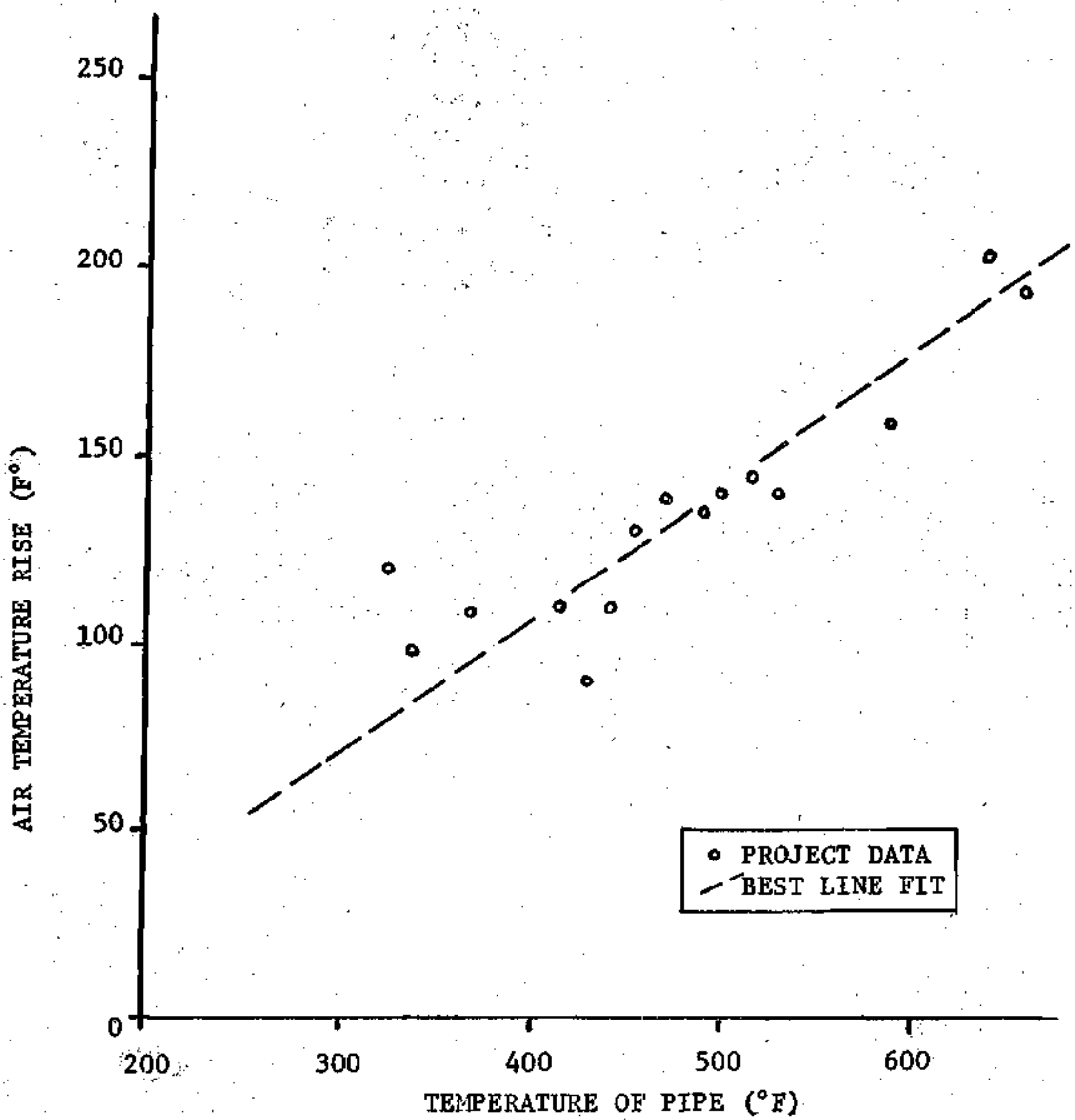

Figure 14. Temperature of Pipe Versus Temperature Rise of Air Through 6 Linear Feet of Heat Exchanger at Flow Rate of 8 cfm (Average Temperature of Inlet Air was 305 °F)

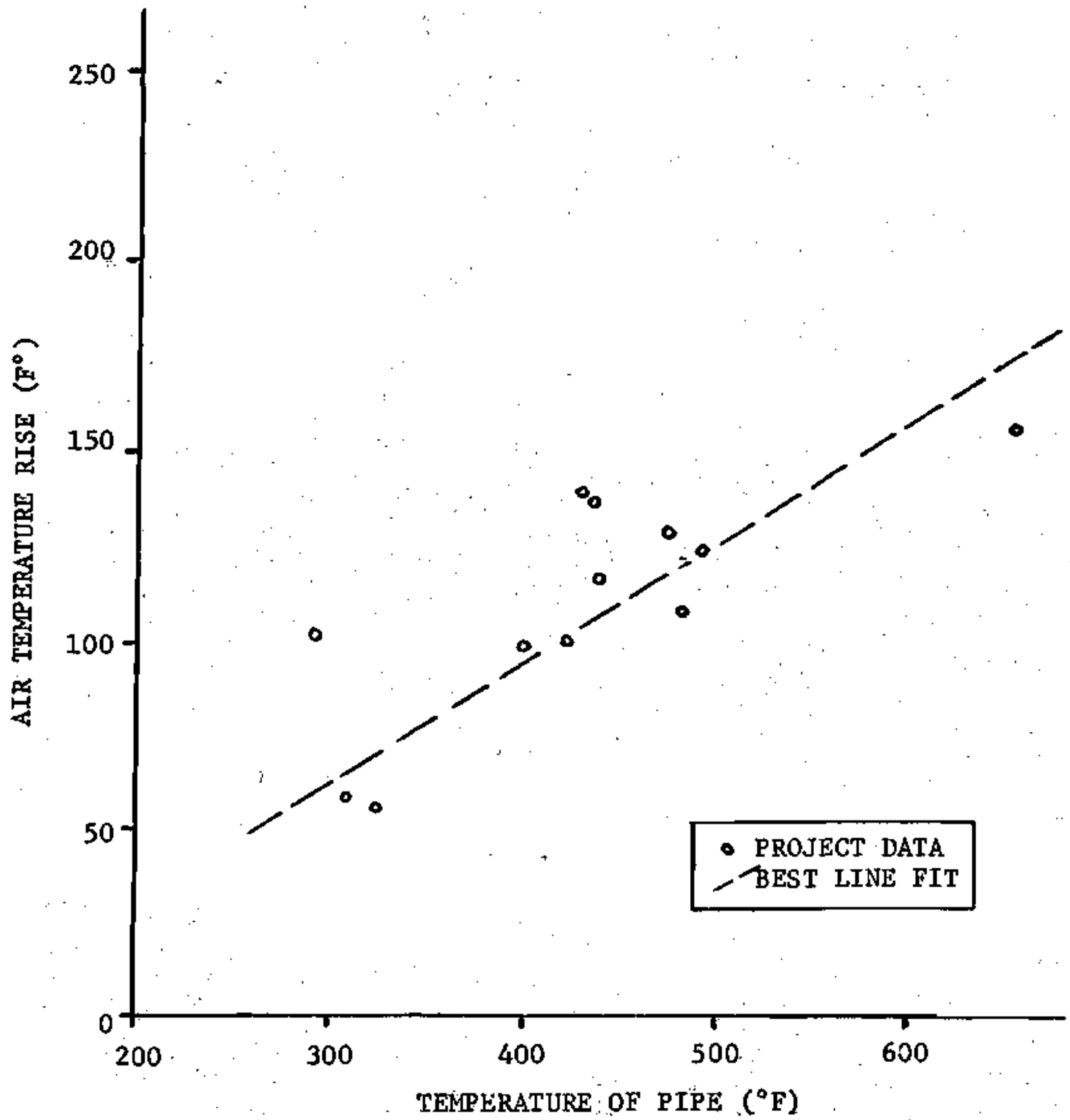

Figure 15. Temperature of Pipe Versus Temperature Rise of Air Through 6 Linear Feet of Heat Exchanger at Flow Rate of 12 cfm (Average Temperature of Inlet Air was 300 °F)

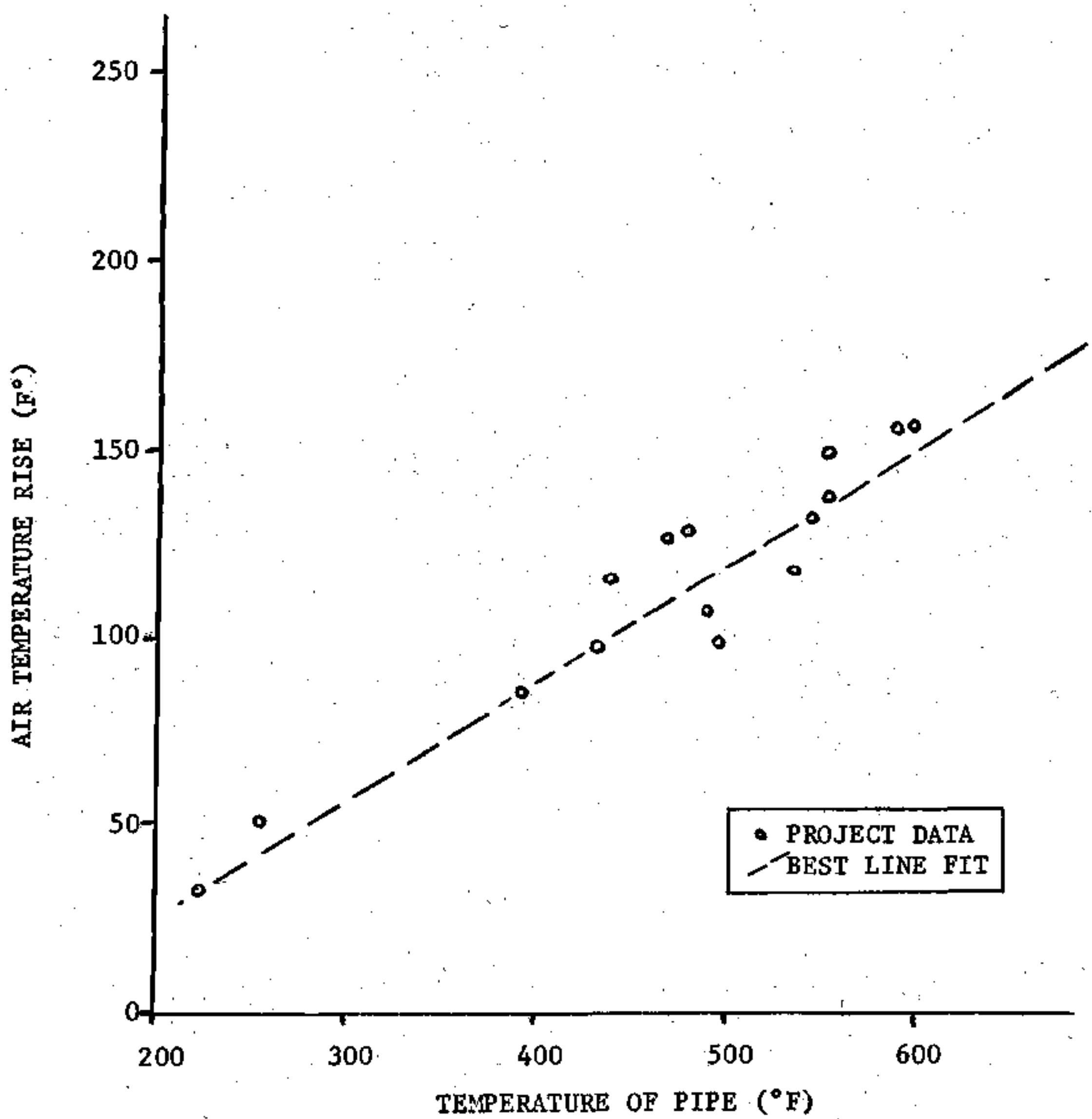

Figure 16. Temperature of Pipe Versus Temperature Rise of Air Through 6 Linear Feet of Heat Exchanger at Flow Rate of 16 cfm (Average Temperature of Inlet Air was 260 °F)

centerline temperature measurements. The only difference this would make is that six feet of heat exchanger would be evaluated rather than 8 feet.

Figures 6-9 show the pipe temperature measurements as a function of the corresponding direct radiation on the concentrator. These radiation measurements were made on a pyrheliometer and recorded as instantaneous readings at two minute intervals. Since it was important that the model be operating at steady state, the pyrheliometer readings taken for the ten minutes prior to the model readings were averaged. This allowed for a ten minute "response time" of the heat exchanger model and would perhaps produce a more accurate value of the direct radiation than would a single instantaneous value.

With Figures 13-16 on hand, efficiency plots in Figure 17 are obtained. As mentioned before, the first-law efficiency is defined as the useful heat input divided by the total heat to reach the heat exchanger. To obtain the useful heat input, q_u, the equation

$$q_u = \dot{m}c_p \rho\Delta T \qquad (12)$$

is used.

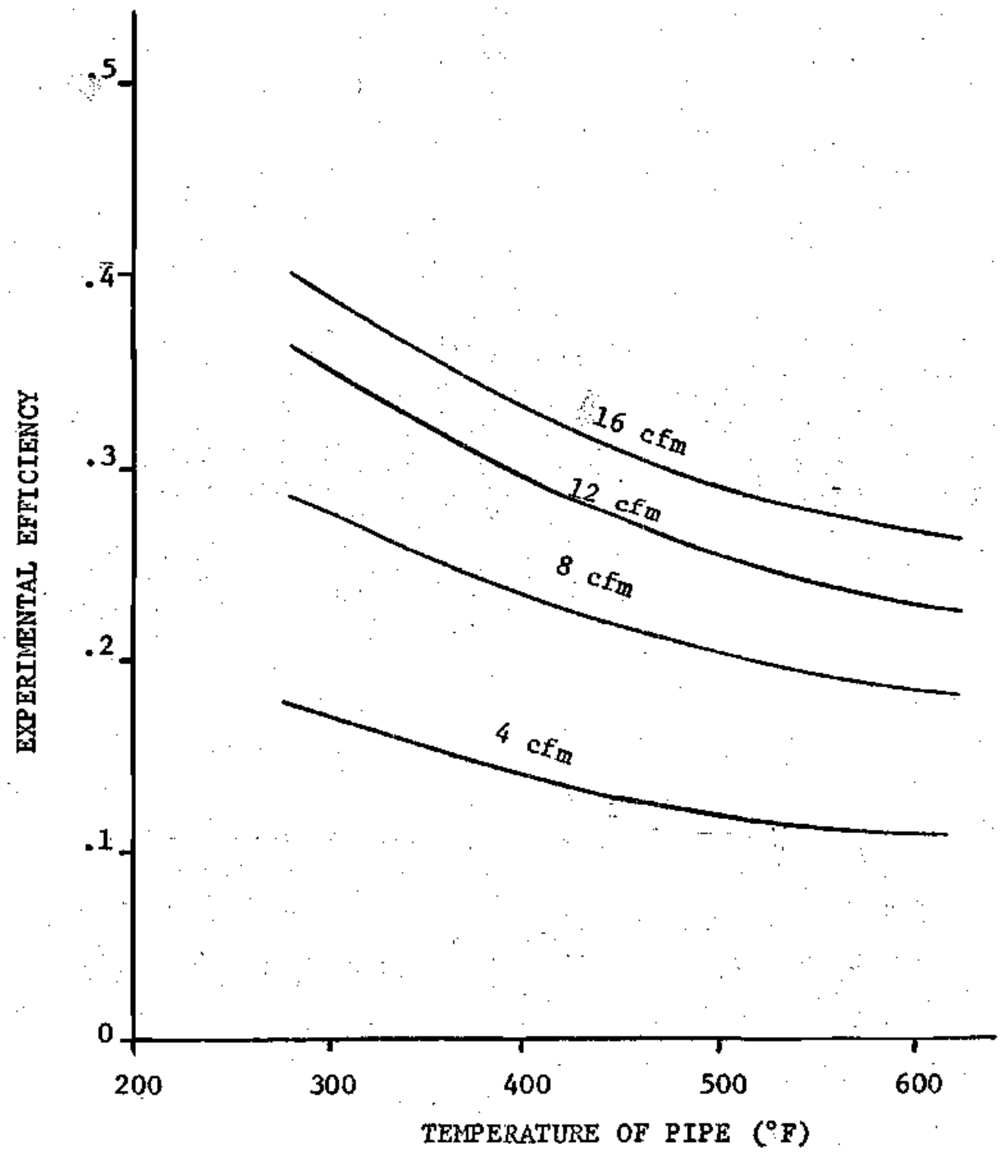

Figure 17. Temperature of Pipe Versus Experimental Efficiency at Various Flow Rates and Wind Velocity of 4 mph

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

The method of designing as well as the construction and the materials did create a working heat exchanger. The method used produced a heat exchanger of low cost, readily available materials and requiring no unusual construction techniques. At the time of the model construction the material cost of the heat exchanger was approximately seven dollars per linear foot.

The experimental performance of the concentrator compares fairly well with the theoretical performance (see Figures 18-21). The main factor contributing toward any difference might be that not all the reflective losses from the mirrors to the heat exchanger were taken into account. In the opinion of this author during the time of testing the dust coating on the mirrors never was of a heavy level, but if the total input value of the radiation onto the test model was lowered, experimental and theoretical values of efficiency would more closely coincide.

One other conclusion worthy of mention is that the test model was located 24 feet from the entrance of the first heat exchanger section. This was done purposely to obtain a fully established flow through the pipe. However, as a result, the temperature of the air entering the test section was not the same over the range of flow rates and a specific pipe temperature. In reality, the useful heat absorption by the air is not linear with length of the heat exchanger so the evaluation of efficiencies

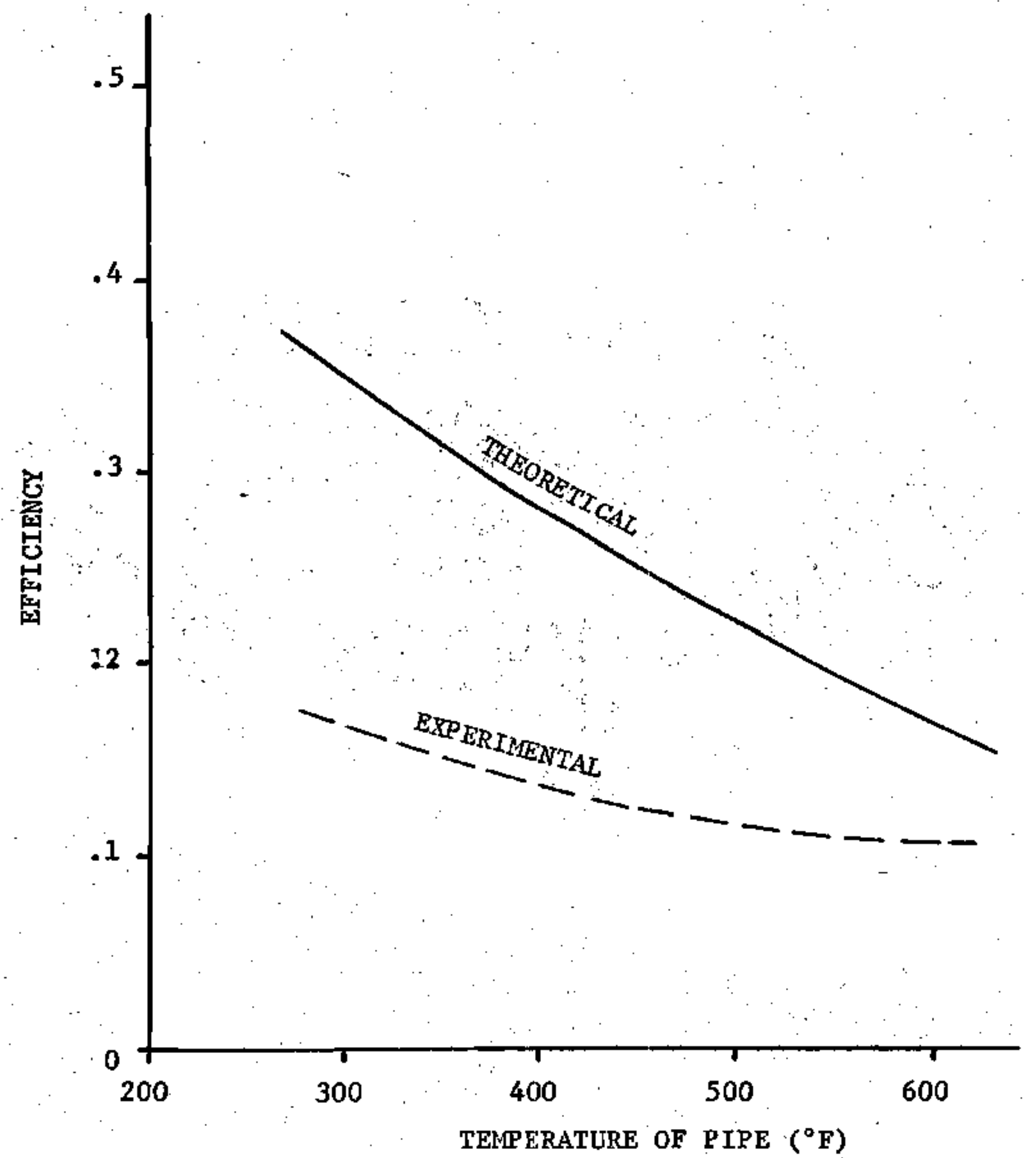

Figure 18. Temperature of Pipe Versus Experimental and Theoretical Efficiencies at Flow Rate of 4 cfm and Wind Velocity of 4 mph

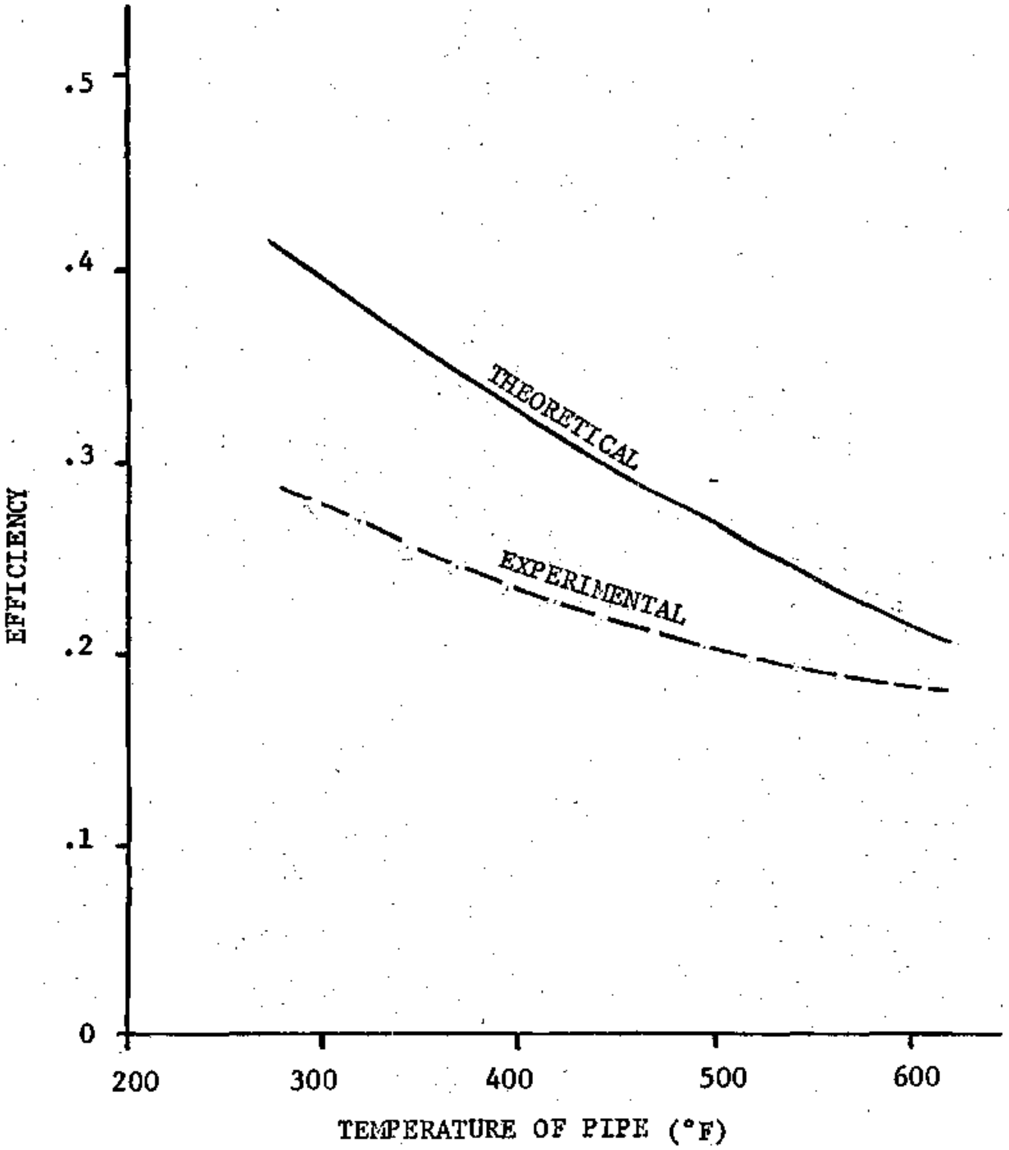

Figure 19. Temperature of Pipe Versus Experimental and Theoretical Efficiencies at Flow Rate of 8 cfm and Wind Velocity of 4 mph

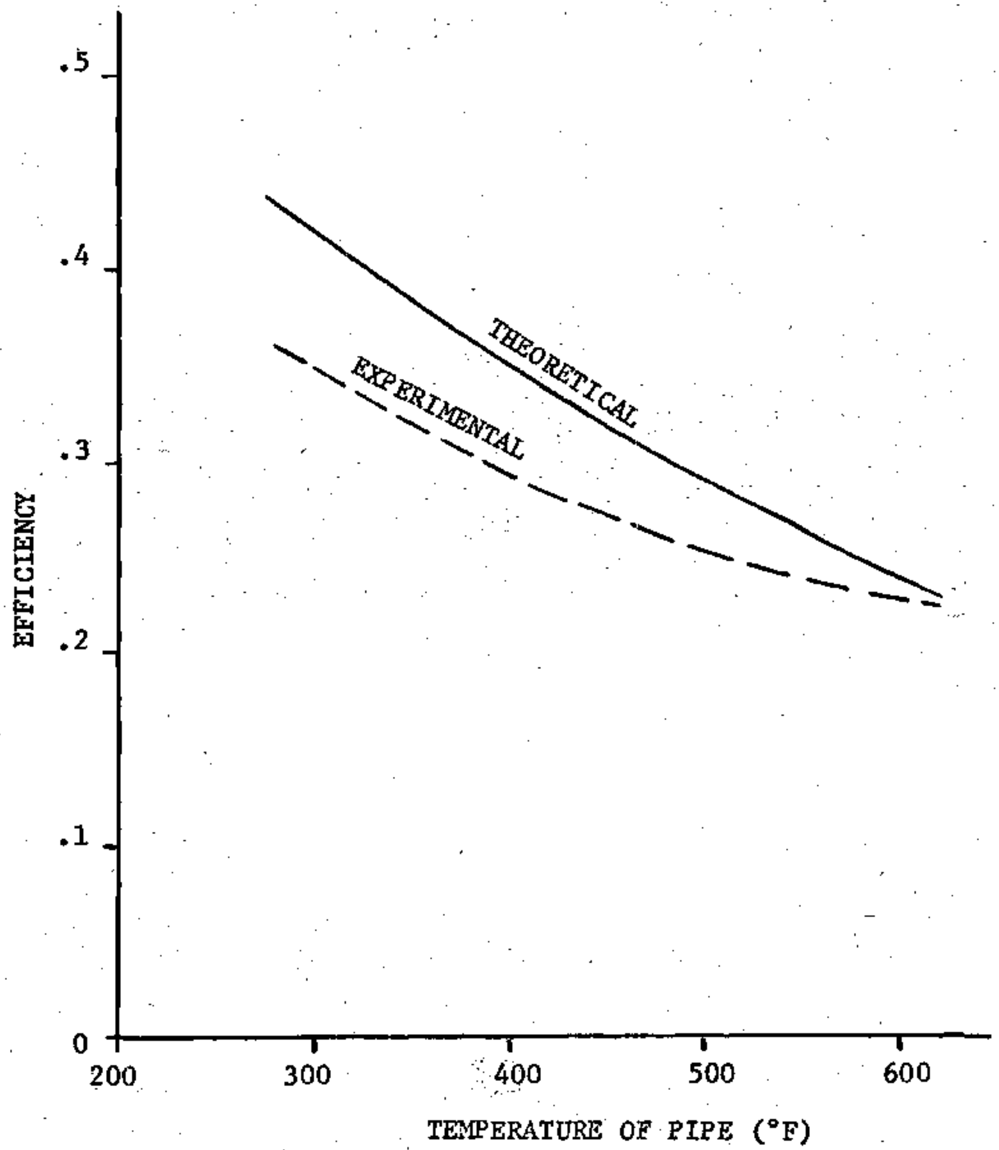

Figure 20. Temperature of Pipe Versus Experimental and Theoretical Efficiencies at Flow Rate of 12 cfm and Wind Velocity at 4 mph

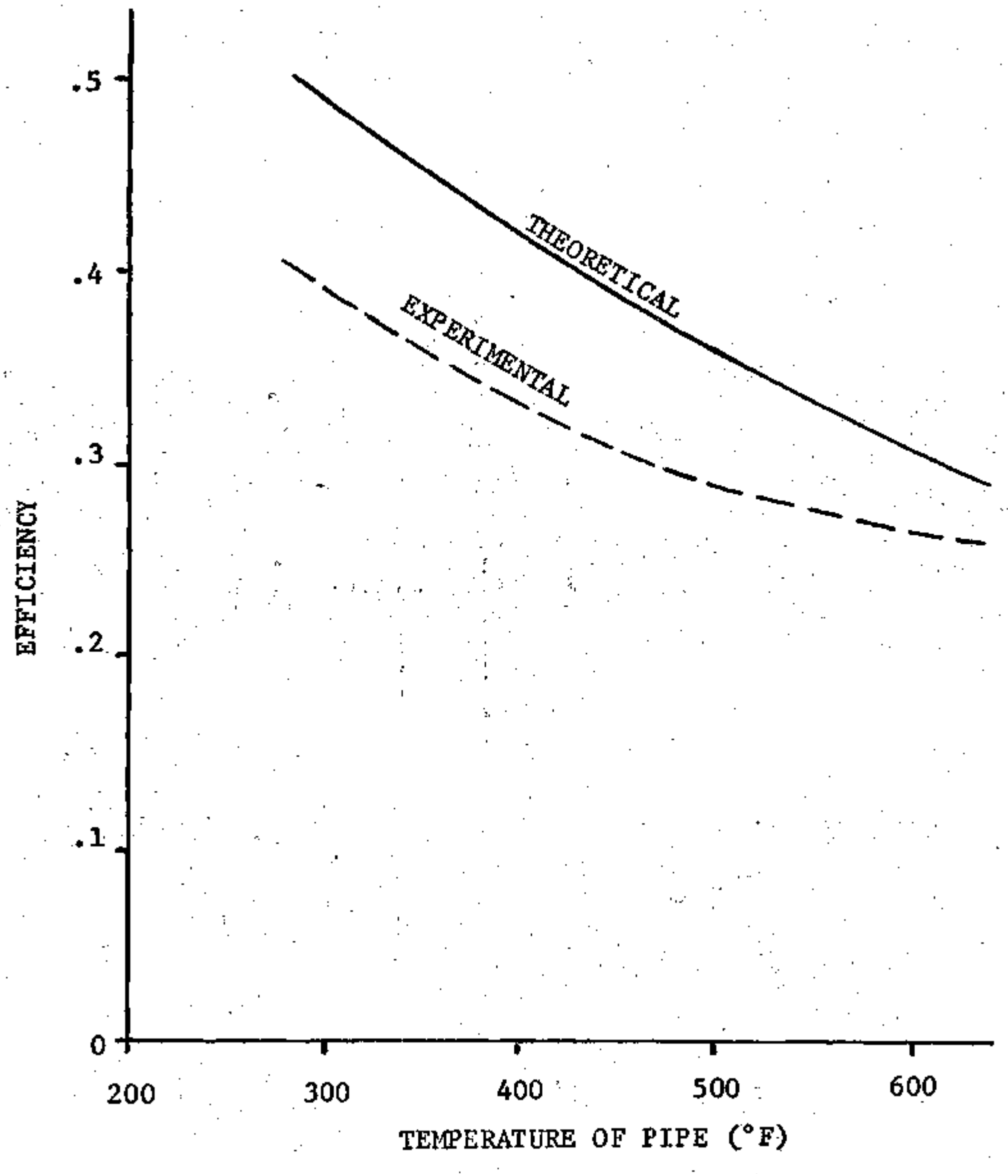

Figure 21. Temperature of Pipe Versus Experimental and Theoretical Efficiencies at Flow Rate of 16 cfm and Wind Velocity of 4 mph

between flow rates should take into account the entrance air temperature.

Recommendations

At no given time was the temperature at station 3 higher than at station 11 thus indicating that at no given cross-section was the air temperature higher than the temperature of the back of the pipe. However, temperature measurements at stations 7 and 11 often differed by 150 °F. This difference might be lowered if a thicker walled inner pipe was employed rather than the pipe that was used. As a result of the consequent more even distribution of heat about the circumference of the pipe better heat transfer from the pipe to the air would be obtained. A finned tube may aid the same heat transfer by providing more heated surface area for air contact.

Another recommendation toward the design of the model is the increasing of air turbulence. This could be accomplished by placing a ribbon of metallic foil down the tube and then leaving it in a twisted position. Another method may be to place a metallic mesh down the tube (after considering the resulting pressure drop). As a result of increased air turbulence there would be more air contact to tube wall thus providing increased heat transfer.

The only recommendation that needs to be made toward the construction of the heat exchanger is that careful attention should be given to air leaks. Steel stove pipes interlock nicely to couple sections but leaks may still occur. Another source of leaks may be the small screw holes made to fasten the terminal concentrator onto the pipe sides. Such leaks during testing may lead to errors in true flow rate measurements.

This problem may be remedied by the application of a high temperature sealant, such as silicone rubber, over and around possible air leaks.

In recording experimental measurements another recommendation might be that the experimenter take care that data is at steady state and carefully noting variances in wind speed and allowing time for the test model to attain steady state operation. Since the sun's radiation is rarely at a steady level pyrheliometer measurements taken on strip-chart recorders may make numerical averaging of sun values more accurate. The same argument may also be made for wind measurements.

Radiation effects from the pipe to the flow stream thermocouples were neglected in this paper. The errors involved in such air temperature measurements if radiation effects are neglected will increase with increasing temperature difference between the pipe and the air. If a heat resistant sheath were placed about the thermocouple in a manner to shield it from pipe radiation the effects could still be neglected and more accurate centerline air temperatures would result.

Finally, it should be pointed out that only the heat exchanger was examined in this report. Before a final decision on the feasibility of the fixed mirror concentrator system is made, the entire system also including the heat storage unit ought to be designed, built and tested. This would provide information on the complete system capabilities and what it would cost.

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